for . . .

engineers and contractors controlling the design, specification, installation and maintenance of heating, ventilating and air conditioning.

HEATING & VENTILATING

AIR CONDITIONING

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MEMBER
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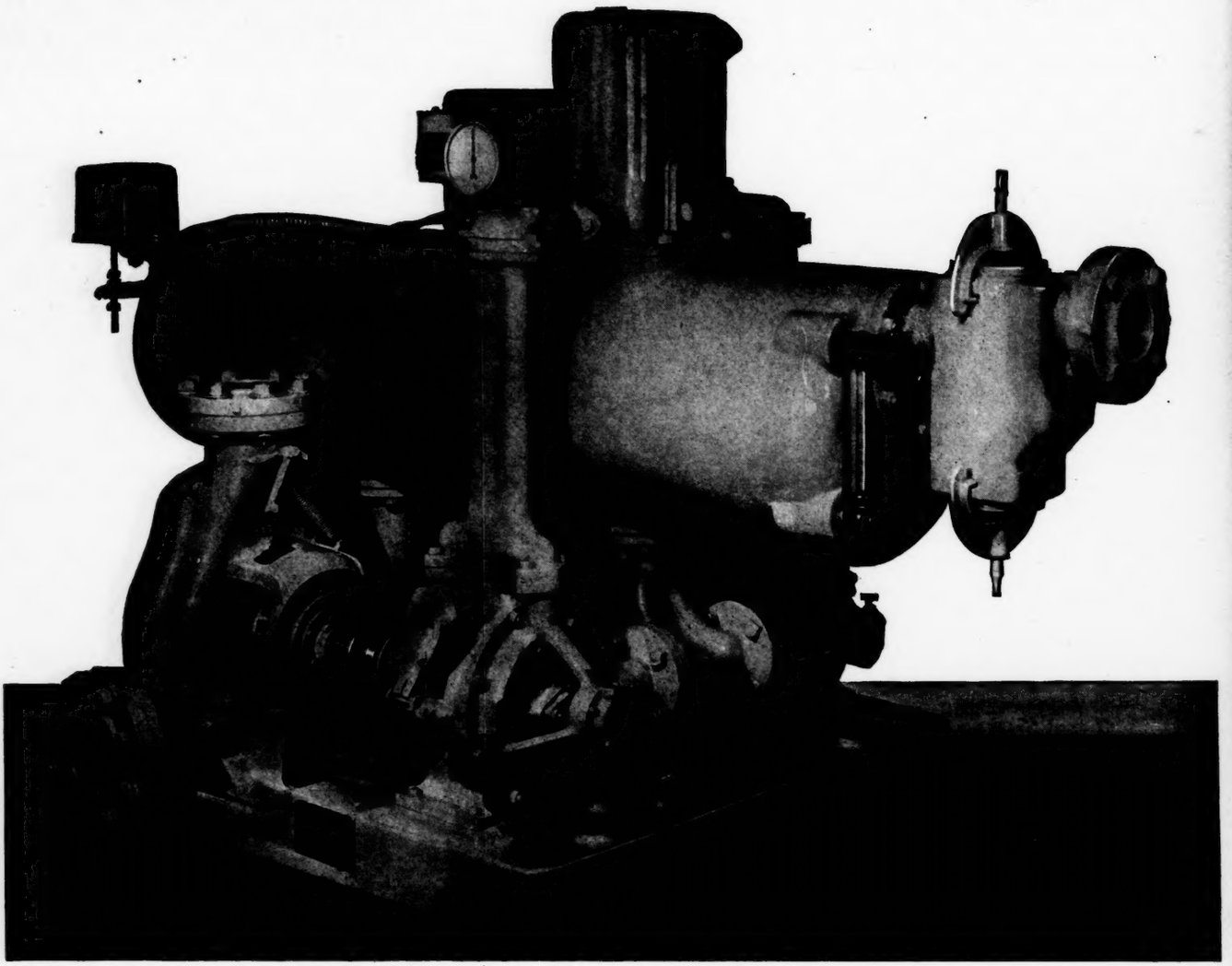
Where heat must *Not* fail . . . install this SAFE *Heating Pump*



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system. This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically by-passes from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure. The Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump.

BULLETIN ON REQUEST

THE NASH ENGINEERING COMPANY
221-AS WILSON ROAD, SOUTH NORWALK, CONNECTICUT

No American now needs to be told that this country has embarked on a tremendous program of building an impregnable system of military and naval defense. Already well under way in this expansion program are dozens of new factories, airports and hangars, naval depots, housing projects for civilian and military personnel, and other buildings involving millions of dollars.

Heating and air conditioning plays an important part in this program. Fully to inform its readers on recent methods and technics employed in industrial building involved in the defense program, HEATING & VENTILATING's September issue will include a large section which will constitute an up-to-the-minute handbook of case histories of defense plant applications of heating and ventilating equipment.

America Builds for Defense

JUST how many of hundreds and millions of dollars in new buildings for industrial plants, airports, arsenals, shipyards, naval stations, barracks, and housing for workers engaged in defense industries will be involved in the American defense program is not known, nor does there seem to be any way to make a close estimate at the present time. The reason for this is, that among other things, it is difficult to separate routine work for various governmental agencies, such as the War and Navy Departments, which was authorized months or perhaps even years ago, and that for which appropriations are being made currently. Additional complications are due to the fact that blanket sums are appropriated for, say, an arsenal, completely equipped with machine tools and other equipment; until the arsenal is designed it is an extremely difficult matter to say just what the building will cost. Still further complications are due to the large number of governmental agencies and sub-divisions involved.

However, at the present time it is not particularly important to determine exactly the amount to be spent for buildings. It is sufficient to know that the amount is enormous. This is illustrated by the proposed program for explosive and chemical plants—certainly only a relatively small part of the total program—as proposed by the Chemical Warfare Service and the Ord-

nance Department insofar as these divisions are concerned with chemical, explosive and related plants. The following table summarizes this proposed plan:

PROPOSED PROGRAM FOR CHEMICAL AND EXPLOSIVE PLANTS FOR THE ARMY

Number of Plants	Type of Plants	Total Cost of Buildings and Land Only
4	Smokeless Powder	$60,000,000
4	TNT	12,000,000
2	Tetryl	2,000,000
2	Picric Acid	2,000,000
2	Cotton Purification	2,000,000
2	Ammonium Nitrate	4,000,000
1	Ammonia	6,000,000
	Chemical Warfare Service	
1	Existing Arsenal	$1,000,000 (est.)
1	New Arsenal	9,000,000 (est.)
26	Commercial Plants	3,000,000 (est.)

Organization for Defense

On July 21st Secretary of War Stimson announced the formation of a construction advisory section of the War Department which will consult with and advise the Quartermaster General, Major General E. B. Gregory, on the $500,000,000 construction program for the War Department. Harry W. Loving, since 1937 executive secretary of the Carolinas branch of the Associated General Contractors of America, has been made head of the Construction Branch of the Construction Division, Army Quartermaster's Corps, under the direction of General C. D. Hartman, chief of the Construction Division, office of the Quartermaster General.

Mr. Loving will be in charge of all defense construction carried out by the War Department on a cost-plus fixed fee basis. Francis Blossom of Sanderson & Porter, New York, Forrest Harvey of Quinton, Code & Hill, Leeds & Bernard, Los Angeles consulting engineers, and J. J. C. Dresser, formerly with PWA as a consultant, have been named to assist Mr. Loving in selecting contractors for the Army construction program.

An Advisory Committee has been appointed by the Army and Navy Munitions Board in connection with the formulation and review of plans pertaining to defense. Civilian engineers, architects and contractors make up the six-man committee which was appointed last May but whose names were not made public until July 22. The committee is headed by John P. Hogan, president, American Society of Civil Engineers, the other members being Alonzo J. Hammond, consulting engineer, Chicago, and president of the American Engineering Council; E. J. Harding, general manager of the Associated General Contractors of America; E. P. Palmer, of the New York contracting firm of Senior & Palmer and past president of the Associated General Contractors of America; Malcolm Pirnie, New York consulting engineer and general chairman of the Construction League of America; and Stephen F. Voorhees, New York architect and past president of the American Institute of Architects. The committee will also work in close cooperation with the National Advisory Defense Commission and particularly with W. H. Harrisson, recently appointed head of the commission's construction division.

From all parts of the country word comes that various groups, industries and individuals are forming organizations and initiating activities, all with the objective of cooperating with the defense program. As announced on these pages last month, a census of 115,000 engineers and architects skilled in construction, is under way under the joint sponsorship of the A.I.A. and A.S.C.E., and the American Society of Heating & Ventilating Engineers is cooperating in this census. The first step in this census is the listing of engineering and architectural firms and individuals in private practice. The second step will consist of indexing and classifying these names according to qualifications. The final step will list and classify individuals and specialists, classifying them according to their ability to do specialized work.

Information on Defense

As might be expected, during the past two months Washington has been besieged by companies and individuals seeking information on industrial mobilization. Those seeking such information include manufacturers, contractors, engineers and others, many of whom have hurried to Washington only to find that they can waste much time in the maze of government offices before reaching the individual or department whom they wish to see. In order to assist those looking for information, the War and Navy Departments, through the Department of Commerce, announced on July 13 that information regarding industrial mobilization can best be secured through the War and Navy Department field offices rather than through Washington. The Navy maintains 27 major field purchasing offices in 27 cities, while the War Department has 46 major field procurement offices in 22 cities. Those seeking information are urged to deal with these local offices who have orders to take the matter up with Washington if the problem is too difficult to handle.

The accompanying table gives the location of these various field offices which are given in some detail, particularly for the Army.

Other sources of information regarding the construction program include the office of the supervising architect in the Department of Treasury, Procurement Division, Public Building Branch; Chief of Construction Division, Office of the Quartermaster General, War Department; Bureau of Yards and Docks, Navy Department; Quartermaster, U. S. Marine Corps., Navy Department; Office of the Chief of the Air Corps., War Department; and United States Housing Authority.

Construction Costs

In the meantime, contrary to what might be expected, construction costs have not risen appreciably. The construction cost index of the American Appraisal Co. indicates that during June there were minor changes both up and down of material and labor, these changes resulting in no change for the average.

NATIONAL DEFENSE PROCUREMENT

The tabulation below, based on information issued by the Department of Commerce July 13, gives the location of field offices of the Army and Navy where information can be obtained regarding the Government's Defense Program. The War and Navy Departments state that current information can best be obtained through these field offices rather than by writing to or sending representatives to Washington. (Also see accompanying text).

Abbreviations below are: CW—Army Chemical Warfare Service Procurement District; AE—Army Engineer Procurement District; MD—Medical Department Procurement District, Army; AO—Army Ordnance District; AOO—Army Ordnance Office; QM—Army Quartermaster Procurement Planning District; SC—Army Signal Corps Procurement District; SO—Supply Officer; OC—Officer-in-Charge.

Alameda, Calif.	—(SO) Naval Air Station
Anacostia, D. C.	—(SO) Naval Air Station
Annapolis, Md.	—(SO) Naval Academy
Atlanta, Ga.	—(QM) 1306 Twenty-two Marietta Bldg.
Birmingham, Ala.	—(AOD) 302 Comer Bldg.
Boston, Mass.	—(CW) 2000 Post Office & Court House Bldg. (AOD) 2004 Post Office & Court House Bldg (QM) Quartermaster Depot, Army Base. (SO) Navy Yard.
Brooklyn, N. Y.	—(MD) 58th St. & First Ave. (QM) 58th St. & First Ave. (SC) 58th St. & First Ave.
Charleston, S. C.	—(SO) Navy Yard
Chicago, Ill.	—(CW) 113 Post Office Bldg. (AE) 1117 Post Office Bldg. (MD) Post Office Bldg. (AOD) 309 W. Jackson Blvd. (QM) 1819 W. Pershing Rd. (SC) 1819 W. Pershing Rd.
Cincinnati, Ohio	—(AOD) 521 Post Office Bldg.
Cleveland, Ohio	—(AOD) 1524 Keith Bldg.
Dahlgren, Va.	—(SO) Navy Proving Ground
Dayton, Ohio	—Army Air Corps, Central Procurement Dist., Wright Field
Detroit, Mich.	—(AOD) 611 Federal Bldg. (QM) 611 Federal Bldg.
Fort Monroe, Va.	—Field Office, Coast Artillery
Great Lakes, Ill.	—(SO) Naval Training Station
Indian Head, Md.	—(SO) Naval Powder Station
Jeffersonville, Ind.	—(QM) 10th St. & Meigs Ave.
Key West, Fla.	—(SO) Naval Station
Lakehurst, N. J.	—(SO) Naval Air Station
Los Angeles, Calif.	—(AOO) 409 Chamber of Commerce Bldg.
Mobile, Ala.	—(AE) 212 Wilson Bldg.
New London, Conn.	—(SO) Submarine Base
Newport, R. I.	—(OC) Navy Purchasing Office
New York, N. Y.	—Army Air Corps Eastern Procurement Dist., 90 Church St. (CW) Room 404—45 Broadway. (AE) 39 Whitehall St. (AOD) Room 1214 — 90 Church St. (OC) Navy Purchasing Office, P. O. Box 9, Station C.
Norfolk, Va.	—(SO) Naval Air Station. (OC) Naval Supply Depot, Naval Operating Base.
Pensacola, Fla.	—(SO) Naval Air Station
Philadelphia, Pa.	—(AE) Second & Chestnut Sts. (AOD) 1417 Mitten Bldg. (QM) 21st & Johnson Sts. (SO) Navy Yard. (SO) Naval Aircraft Factory, Navy Yard.
Pittsburgh, Pa.	—(CW) 1014 New Federal Bldg. (AE) 1012 New Federal Bldg. (AOD) 1032 New Federal Bldg.
Portsmouth, N. H.	—(SO) Navy Yard
Puget Sound, Wash.	—(SO) Navy Yard
Rochester, N. Y.	—(AOD) 1118 Mercantile Bldg.
St. Louis, Mo.	—(MD) 2nd & Arsenal Sts. (AOD) 935 Custom House. (QM) 2nd & Arsenal Sts.
(Fort) Sam Houston, Tex.	—(QM) Quartermaster Depot.
San Diego, Calif.	—(SO) Naval Air Station, North Island. (OC) Naval Supply Depot, Naval Operating Base.
San Francisco, Calif.	—(CW) 117 Federal Office Bldg. (AE) 409 Custom House. (MD) The Presidio. (AOD) 118 Federal Office Bldg. (QM) Fort Mason. (SC) The Presidio. (OC) Navy Purchasing Office.
Santa Monica, Calif.	—Army Air Corps Western Procurement Dist., 506 Santa Monica Blvd.
Springfield, Mass.	—(AOD) 3640 Main St.
Washington, D. C.	—(SO) Navy Yard. (SO) Naval Research Laboratory.
Yorktown, Va.	—(SO) Naval Mine Depot

USHA and Defense Housing

Broadly speaking, the housing which it will be necessary to provide in connection with the defense program will be of three general types. First will be the barracks and cantonments and similar types of structures for housing military and naval personnel on land duty; second, and really a subclassification of the first, is the housing for families of enlisted personnel, and third, will be the additional housing necessary to house workers in defense industries in regions where existing housing facilities are insufficent.

As a first move to meet housing needs in connection with the second two classifications, President Roosevelt late in June approved loan contracts for local housing authorities in Montgomery, Ala., and Pensacola, Fla., providing USHA loans of approximately $2,000,000. The project at Montgomery will consist of 424 dwelling units to house enlisted personnel and civilian employees at Maxwell Field, an Army air base. Two hundred dwelling units will be built at Pensacola to house personnel engaged in activities at Corry Field, a naval air station at Pensacola. Both of these projects will consist primarily of one and two-story row houses. Each dwelling will have hot water heating facilities.

By July 3, USHA Administrator Nathan Strauss had earmarked $3,500,000 in USHA funds for eight other defense housing projects in Beaver County, Pa., Chester County, Pa., Erie, Pa., McKeesport, Pa., Pittsburgh, Pa., Newport News, Va., Corpus Christi, Tex., and Portsmouth, Va. The project at Portsmouth is designed for housing workers at the Navy Yard and ammunition depot there; that at Newport will provide homes for the workers at the U. S. Naval Station and a torpedo manufacturing plant there; that at Corpus Christi is for the huge naval base in that locality which is not yet completed. The other projects are for workers engaged in essential industries, particularly steel, in western Pennsylvania, excepting that of Chester County in eastern Pennsylvania.

About 450 local housing authorities will be enlisted in the defense housing program, as a result of a bill H.R. 9822 (to expedite naval ship building and other purposes), recently passed by Congress. In this bill, USHA and the War and Navy Departments are authorized to cooperate in making housing available for persons engaged in national defense activities including all of the three types outlined previously. According to a conference report on the bill, recruitment of personnel is already being hampered by lack of housing in many places, including Charleston, S. C., Vallejo, Calif., the Panama Canal, and Hawaii.

Participation of USHA in the defense program and the use of this already well organized governmental agency for aiding defense housing was expedited by a decision announced by the Supreme Court of Texas last month declaring the validity and constitutionality of the Texas Housing Authority's law which permits Texas communities to participate in the slum clearance and low rent housing program of the authority. This decision was the thirty-fourth decision so favoring local housing authorities by the states. Only ten states now lack state laws enabling cooperation with USHA. They are Iowa, Kansas, Maine, Minnesota, Nevada, New Hampshire, Oklahoma, South Dakota, Utah and Wyoming.

Financing of Buildings

Where buildings are to be erected purely for any governmental agency or department, such as the War and Navy Departments, the financing is, of course, a governmental function and a governmental problem. It is expected that housing needs will be taken care of as far as possible through USHA where such housing is of the conventional type, and local housing authorities and their facilities will be employed. It is probable that USHA's current practice of employing private engineers, architects and contractors in the individual contracts will be followed. Housing of the barracks and cantonment type, however, will probably fall into the category of work done directly for the government and will be paid for by the government department involved.

The National Defense Advisory Commission, appointed by the President some months ago and headed by William S. Knudsen, is apparently disposed to encourage production in privately owned and operated plants. On July 25, the Treasury approved a method by which corporations undertaking defense contracts may, over a short period of time, charge off to depreciation the cost of new plant and equipment necessary for defense orders, Secretary Morgenthau said.

The statement came as Congressional leaders prepared to introduce legislation to combine an excess-profits tax with a provision to permit writing off depreciation on new defense plants in five years.

The legislation was approved in principle on July 10 at a White House conference attended by President Roosevelt, Secretary Morgenthau, Congressional leaders, William S. Knudsen of the National Defense Advisory Commission and Jesse H. Jones, Federal Loan Administrator.

The loophole in existing law which is the basis for the new ruling by the Treasury may serve as a substitute for such legislation as far as many corporations are concerned. However, the proposed bill is expected to be moved for passage anyhow.

The only important stipulation regarding the quick write-off is that the new plant and equipment needed to fill defense orders must be built on leased land. Contractors, however, would be permitted to set up a dummy corporation to take title to the proposed factory site and lease it to the operating company.

Fly in the Ointment

However, in spite of the large volume of building work under way and contemplated in connection with the defense program, all is not quite serene so far as certain groups in the building industry are concerned. There is some evidence of discontent on the part of both the professional and trade groups engaged in heating due to the methods of handling building construction projects by the Navy Department and with indications that a similar practice may be followed by the War Department in the future.

The traditional method of handling heating contracts on large scale government projects has been for a

private architect, consulting engineer, or architectural and engineering department of a government bureau to design and draw up specifications of the building, which was then constructed by a general contractor, with the heating handled as a sub-contract. Beginning early this year, however, the Navy Department began to let the contracts under a cost-plus-fixed-fee basis whereby the basic drawings are prepared by private architects and engineers, and arrangements made with the general contractor to prepare working drawings, and to construct the building, including work traditionally handled by subcontracting, all on a cost basis plus a negotiated fee. The fee represents the general contractor's profit.

Two different groups in the building field are seriously concerned with this method of handling contracts. The first group consists of the private architects and consulting engineers who feel that the plans and specifications should be carried all the way through by private professional firms. The second group is composed of the subcontractors (such as heating, plumbing, and electrical) who fear that they are or will be frozen out by the general contractor under this method.

On the other hand the governmental agencies concerned are necessarily laboring under a heavy burden in pushing through an extraordinarily large volume of construction during the emergency period and, consequently, wish to operate under a method which will permit the erection of buildings in the most expeditious manner.

Nevertheless, the situation may result in a tendency for the general contractor to set up his own architectural, engineering and subcontracting departments so that the cost involving these phases of work would be paid entirely by the government as part of the cost of the project with the entire fee retained by the general contractor.

Representations have been made to both the Navy and War Departments by the Heating, Piping and Air Conditioning Manufacturers Association; National Association of Master Plumbers, and National Electrical Contractors Association, who claim that the government's method results in increased costs, increased time, and lower quality of work than existing subcontractors are capable of turning out.

However, on July 20 Rear Admiral B. Moreell, Chief of the Bureau of Yards and Docks of the Navy Department, issued a memorandum to the commanding officers of naval stations, naval air stations, and naval districts, various naval officers in charge of construction, as well as to numerous large general contractors, which included two paragraphs as follows:

"In general, it is contemplated that those specialties which are usually subcontracted by a general contractor may be subcontracted on the fee contracts, and this is taken into account when the fee is fixed. Under such circumstances the subcontract cost is considered as a material purchase and is fully reimbursable as a cost under the contract.

"It is the Bureau's intention that every advantage shall be taken of the availability of skilled specialists for subcontract work, but under some circumstances—such, for example, as at an isolated station where transportation, housing, recreation, etc., are serious problems—it may be impracticable to use subcontracts."

The policy at both the War and Navy Departments apparently is that subcontractors will be used only when it is to the government's advantage to do so.

Representative architects and engineers have already made known their opposition to the present plan and have pointed out to the departments involved that if the cost-plus-fixed-fee basis method is followed through, almost every engineer or architect will be affected, so that the memberships of the American Society of Heating and Ventilating Engineers, American Society of Refrigerating Engineers, American Institute of Architects, American Society of Mechanical Engineers, American Society of Civil Engineers, American Institute of Consulting Engineers, American Society of Professonal Engineers and the American Institute of Electrical Engineers, as well as a number of other engineering societies, will be very much concerned.

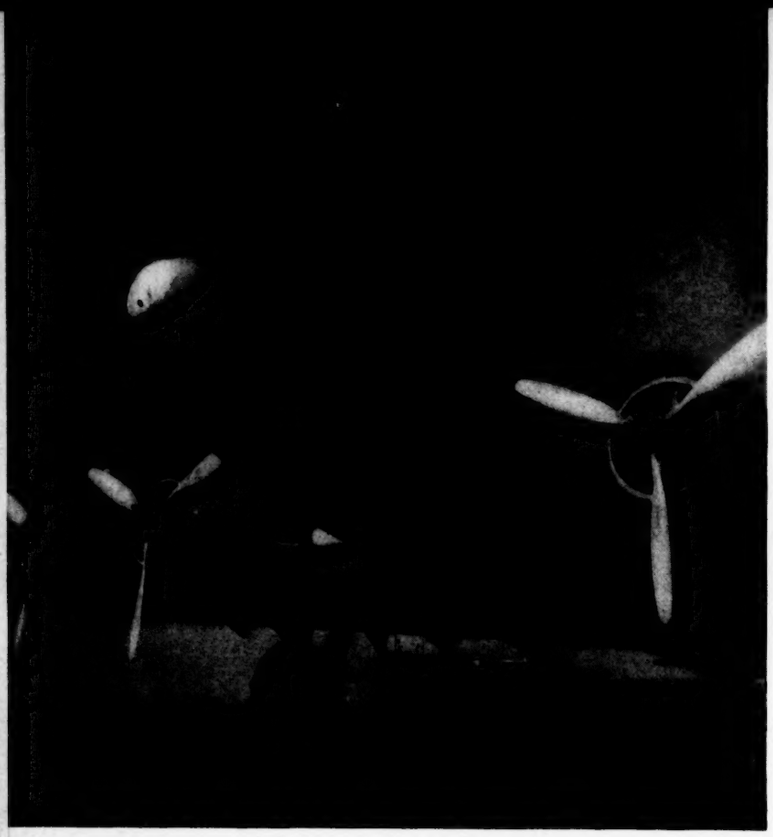
So far as architects and engineers are concerned, however, Admiral Moreell, in a letter to HEATING & VENTILATING July 25, states that "With respect to outside engineers and architects, the instances where the general contractor furnishes the engineering services for our work are very few. In fact, offhand I can recall only one case and in this instance the great bulk of the engineering and architectural work is being furnished by a consulting architect and his organization directly for the Navy Department."

In connection with the general subject Admiral Moreell further pointed out that "In the ultimate analysis it would be proper for all of us to remember that the objective of this program is to prepare this country to defend itself, and not to assure profits to any particular group or groups of citizens."

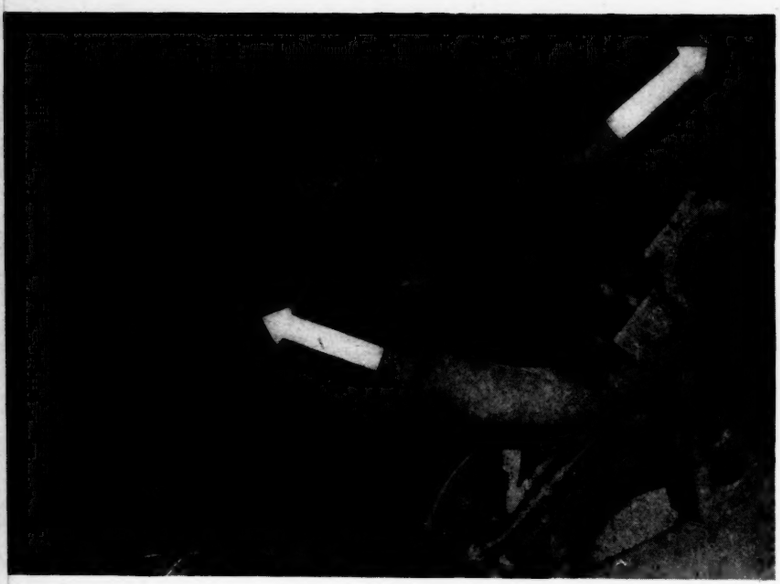
At the time of going to press it is difficult to understand just what the situation is. It certainly is rather confusing. Certainly no architect, engineer or contractor wants to be in a position of in any way blocking or slowing up the defense program. On the other hand there seems to be no good reason why the general contracting group should be the only one out of the building industry to participate in the activity resulting from the defense program. Working through the general contractor to the extent which the method outlined involves may be efficient from the standpoint of a governmental department alone, but it is by no means clear that in the final analysis such a method will be more efficient; such a system may be a great deal more expensive than to follow the traditional method working through existing and smoothly functioning organizations already set up and doing business.

It is believed that one of the reasons for the adoption of this system of handling building contracts by the Navy Department was to simplify procedure in that, by this method, one firm is responsible for the whole project. It is expected, however, that when the various aspects of the matter have been fully investigated by the War and Navy Departments amendments will be made so that the professional and sub-contracting groups, as well as the general contractors, will be allowed to cooperate fully in the defense program.

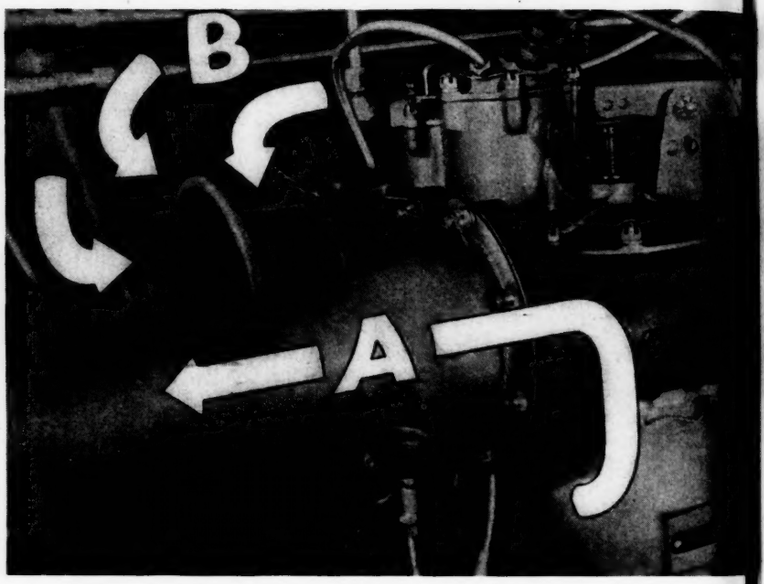
Altitude Conditioning



A new air conditioning function—the maintenance of air pressure has been added to the commonly accepted four functions of air conditioning—control of temperature, moisture, movement, and cleanliness—in the conditioning systems of these new Boeing Stratoliners. Although these planes will fly at altitudes of 15,000 to 20,000 feet, the passengers will not be uncomfortable in the decreased air pressure as air in these planes can be carried at a pressure of 6 lb. per sq. in. higher than the outside pressure. In the picture at the left an air intake duct can be seen in the leading edge of the wing between the nacelle and the fuselage on the right.

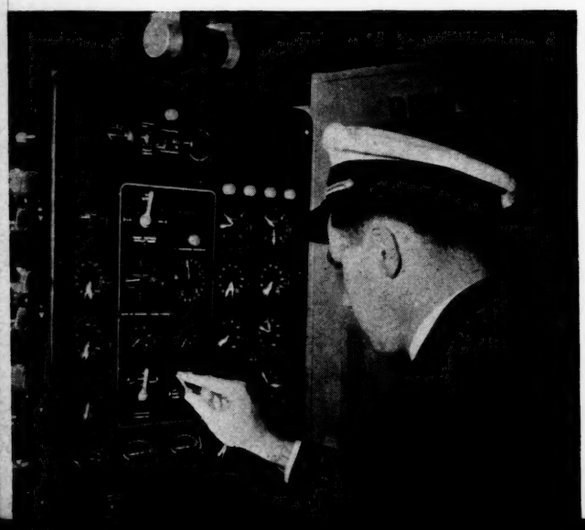


Air from the intake duct enters supercharging unit which is driven from the engine. Arrows show ducts which conduct the air from the wing air inlet to the supercharging unit and thence to the automatic pressure controlling unit.



Air enters the automatic control unit A from the supercharger. This unit automatically maintains low altitude pressure in the cabin when the plane climbs to higher altitudes. B shows outlet which exhausts air from the cabin of the plane to outside.

Flight engineer's station in the control room. The instruments within the white line are for the control of the cabin's duplicate air supercharging system.



One of the ceiling grilles which supply air to the cabin which seats 33 passengers. There are four six-passenger compartments and nine individual reclining chairs.

Main aisle of the four-engine transport. At right can be seen the individual ventilators at each seat which are under passengers' control.

—The Fifth Air Conditioning Function

OPENING a new era in American air transportation, a fleet of five four-engined Stratoliners have recently been finished for Transcontinental & Western Air Lines, and two of a slightly different model have been delivered by Boeing Aircraft Company of Seattle to Pan American Airways for international service between the United States and South American cities. These two groups of planes mark the beginning of the 4-engine era in United States air transportation.

Although flying at 15,000 to 20,000 feet or more above the earth, passengers in these new stratosphere airplanes will not have to worry about lack of oxygen; and the towering Andes below them will look like a range of foothills.

The thin air at these stratosphere levels will be scooped in, compressed to the conditions at about eight thousand feet, or normal flying heights, and delivered to passengers and crew as the plane speeds through the upper regions. Non-stop flights will be made from Brownsville, Texas, to Rio de Janeiro in twenty hours and to other South American points in proportionate time. In the altitude range between 14,000 and 20,000 feet, pilots of these planes can override most earth-surface conditions, have a wide margin of altitude and find smoother air in which to fly, with steadier winds and at higher speeds.

The planes are built to withstand supercharging at six pounds per square inch, but will normally operate at approximately 2½ pounds which is the differential between outside atmospheric pressure at the high altitude and inside pressure at ordinary flying altitude.

The Stratoliner is an all-metal, low-wing monoplane, and its design, distinct from all previous transport planes, is in the form of an elongated teardrop, completely streamlined, perfectly circular in cross-section from nose to tail. Passenger capacity is 33 persons by day or 25 at night (sixteen berths plus 9 reclining chairs). Cargo capacity is 6590 pounds; fuel capacity, 1700 gallons. Each has a crew of five, including captain, first officer, flight engineer and two hostesses.

These airliners are not only supercharged under pressure to ensure proper atmospheric conditions for passengers and operators, but are also fully heated and ventilated, and the planes can be operated either with or without supercharging at any altitude. For lower altitudes, when hot weather frequently prevails, a separate cold air system has been installed in addition to the supercharging system.

At the top of the cabin nose is a *scoop* into which a blast of fresh air is drawn and then passed through a centrifugal rain separator into a central duct in the cabin ceiling. From here it is distributed through grilles. There is an auxiliary ventilation system in the bottom of the accessory apartment below deck from which a large quantity of cool ventilation air at low flight levels can sweep through the passenger compartment to be discharged later to the outside. This is adjustable to weather conditions.

Beside the overhead grilles, air from the ceiling duct is distributed direct to passengers in their seats or berths, by means of individual ventilators, with nozzles having volume and direction flow by means of a ball and socket device, easily operated. Ventilators have a rounded air inlet and smooth throat and are made of sound deadening plastic material, and provide a high velocity air jet, silent in operation, with a maximum of air recirculation effect.

In cold weather the overhead fresh air scoop can be closed and warm air substituted in the cabin ventilation through the main supercharging and air heating systems, with vent through the auxiliary ventilation outlet. This is another form of non-supercharging ventilation.

At 8,000 feet and above all ventilation is carried through the supercharging system, with automatic pressure control and automatic temperature regulators. From 8,000 to 14,700 there is no change in pressure. Above the higher altitude, the differential of 2½ pounds per square inch comes in, so that the *apparent* altitude, that is, the altitude which the passenger feels, gradually increases until at 16,000 feet actual altitude, the passenger feels an apparent altitude of 9,000 feet. At 18,000 feet this *apparent* altitude goes to 10,600 feet, and at 20,000 feet it is at approximately 12,300 feet, or about four miles.

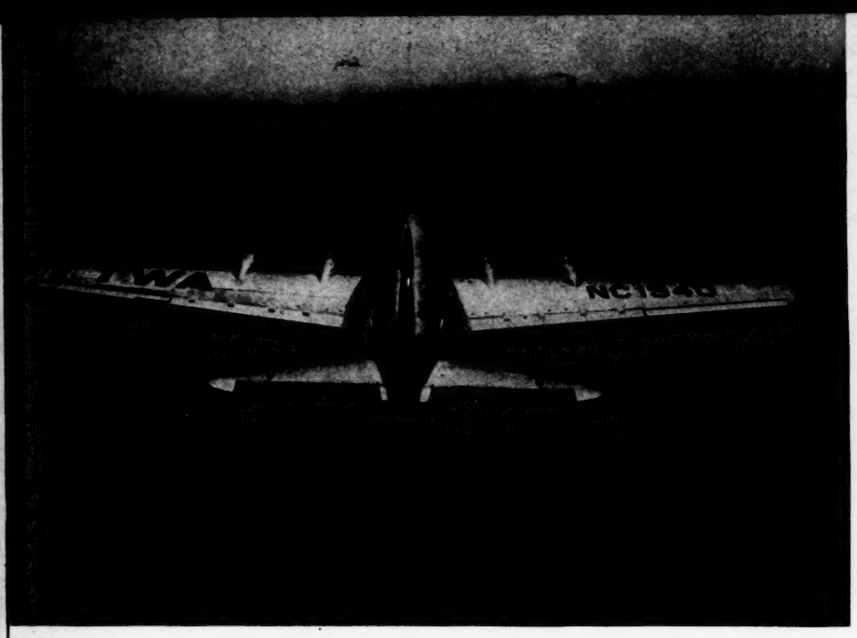
The Supercharging System

There are duplicate supercharging systems, either of which may be used at any time should the other fail. Fresh air is drawn through an opening in the lower leading edge of the wing between the inboard engine nacelle and the fuselage or body. Passing through a centrifugal water separator to the supercharger unit behind the engine of the inboard nacelle. The air is here compressed, warmed by steam radiators, and then flows through a duct to the automatic control valve in the accessory compartment below deck, from which a constant weight flow of air is maintained to all parts of the cabin.

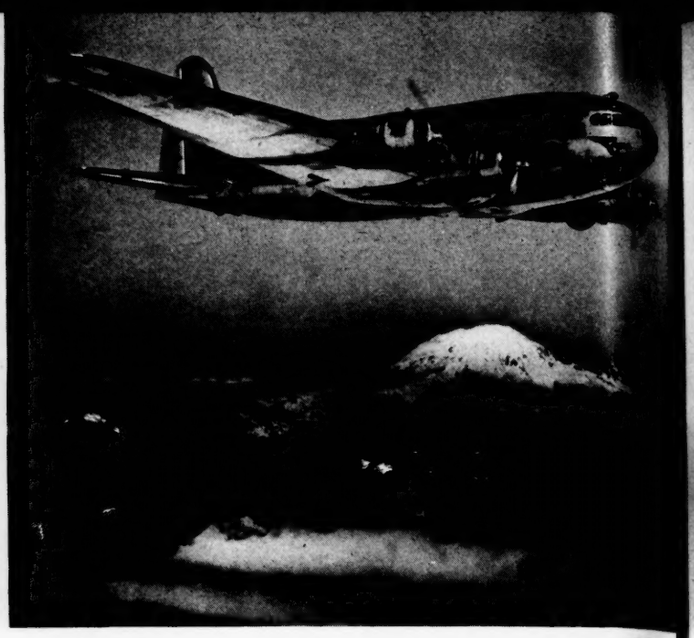
After circulation, the air escapes through mushroom-shaped outlets to the accessory compartment and is passed through the automatic pressure control valve to the outside. Pressure control valve and inflow control valve are combined in one unit, called the cabin supercharging control unit. There are two of these, one for each of the duplicate systems.

Should the air inlets become clogged, a secondary inlet valve opens automatically to furnish an air supply for the supercharger blower.

In the duct from the blower to the cabin is a blower relief or surge valve which opens automatically by the pressure built up in the duct when the air flow in the cabin is throttled down by the valve in the control unit. This bypasses the air from the supercharger and prevents overheating of the latter. During low altitude



This unusual picture of the new B-307 Stratoliner for TWA indicates the size of the streamlined fuselage which accommodates 32 passengers. The fuselage is completely circular in cross-section from nose to tail—the ideal type for supercharging. The wing spread of the big airplane is 107 feet.



One of the Stratoliners on a final check flight over Mt. Ranier. Although Mt. Ranier towers 14,408 ft. into the air, the four-engine Stratoliner can carry its 33 passengers in altitude conditioning comfort at altitudes up to 20,000 ft. It is the world's first cabin supercharged commercial transport plane.

flying in warm weather the entire air output of the superchargers may be bypassed through these relief valves by shutting off the control unit airflow valve. Ventilation then comes from the auxiliary cold air system.

Heating the Supercharged Air

A single steam heating system in the engine nacelle warms the air as it leaves the supercharger, steam being generated by a boiler consisting of a finned cylinder that extracts the heat from the engine exhaust, the steam passing into banks of condenser tubes which warm the air as it passes them. Water for the boiler is contained in a conical sump below the boiler, the sump so designed that should the water freeze when the engine is idle, starting of the engines will immediately thaw it. A feedwater pump, driven by oil from the airplane's hydraulic system, circulates the water from the sump to the boiler. A thermostat, whose setting can be varied at will by the airplane steward or hostess, automatically controls the speed of the feedwater pump, which in turn by its speed, regulates the heat output. There are also optional flow courses for the ventilating air, according to hot or cold weather, directed by the steward or hostess.

Many months of development and experimental work are represented in the compact supercharging control unit which automatically regulates the atmospheric pressure level of the cabin. Frosting of the air outlet is prevented by locating the pipe through which air is discharged to the outside directly through the duct for incoming warm air. All supercharging, heating, ventilating and pressure regulating apparatus is in duplicate, and operation of the system is in charge of the Flight Engineer, who has an instrument board with a compact group of pressure-cabin instruments. A feature of the Stratoliner is the accessibility of controls and equipment and ease of maintenance, with accessibility from an outside door in the accessory compartment, and another reaching the control cables and lines by way of a door in the forward cargo compartment, both reachable during flight.

Control of Foundry Dust Hazard

Although good housekeeping (in foundries) is an important part of a dust control program, nothing short of exhaust ventilation will completely eliminate the occupational disease hazard attending the shake-out, core removal, arbor pulling, tumbling, grinding, sand-blasting, and sand conditioning in mechanical equipment.

In North Carolina the exhaust ventilation of grinding wheels and tumbling mills has been found to effect a 94% reduction in the dust exposure of the operators. These two operations, together with casting shake-out, are the principal sources of dust in the small foundry; however, due to the character of the dust, the shake-out exposure in equal concentrations is not as hazardous as the others. The shake-out exposure in the small plants must be controlled partially by good housekeeping.

The modern mechanized foundry is most amenable to the control of dust by exhaust ventilation. In such plants flask dumping mechanisms, the rotary shake-outs, the tumbling mills, the grinding wheels, and the sand conditioning system all may be readily ventilated, with resulting complete control of the dust hazard.—*M. F. Trice, Industrial Hygienist, Div. of Industrial Hygiene North Carolina State Board of Health, in the "American Journal of Public Health."*

Heating Coil Selection by the By-Pass Method

By C. O. MACKEY† and L. T. WRIGHT, JR.‡

IT has long been customary to select coils for heating, reheating, or tempering service by the use of extensive tables prepared by the manufacturers of the various coils. While these tables give all of the necessary data for the selection of coils, they do not present the data in as graphic a form as might be desired. It is possible by the use of the *by-pass factor* concept to materially simplify coil selection as well as to reduce the space required for presenting coil performance data.

The contact-mixture principle, originally proposed by W. H. Carrier[1] gives, by analogy, a simple explanation of the changes of state of air during many conditioning processes. The assumptions made in the contact-mixture theory are, that in the transfers of heat and water vapor, (1) particles of an air stream come in contact with a hypothetical surface, (2) the state of the *contacted* air particles is that of *saturated* air at the temperature of the surface, (3) there is the equivalent of perfect contact of a definite portion of the air particles with the surface and no contact, or an equivalent by-pass, of the remaining particles, and (4) the end state of the air is the same as that produced by mixing of the *contacted* and *uncontacted* particles.

For a general mathematical deduction from this theory, assume that a fluid passes over a surface of area, A, with an equal number of fluid particles, m, contacting each unit of surface in unit time. The fluid is assumed to consist, at any section, of a homogeneous mixture of contacted and uncontacted particles (this fluid would be completely turbulent). Let Z be a measure of the total number of particles of the fluid passing each section in a unit of time, and let Y be a measure of the number of contacted particles passing the section in unit time. The increase in the number of *contacted* particles per unit of time is to the total number of contacts per unit time as the number of uncontacted particles in the mixture passing the section is to the total number, or

$$\frac{dY}{mdA} = \frac{Z-Y}{Z} \text{ or } \frac{m}{Z}dA = \frac{dY}{Z-Y} \quad \ldots\ldots (1)$$

To integrate this differential equation, assume (1) that the same total number of particles pass each section in unit time, (2) that the total number of contacts with each unit of surface in unit time is constant, and (3) that there are no initially contacted particles; then

$$\frac{m}{Z}A = -1n\frac{Z-Y}{Z} \text{ or } \frac{Z-Y}{Z} = e^{-\frac{mA}{Z}} = X \quad \ldots\ldots (2)$$

The ratio, $\frac{Z-Y}{Z}$, may be called the equivalent by-pass factor, X, since this ratio represents the fraction of the total fluid that may be considered to have by-passed the surface if the remainder of the fluid may be assumed to have made perfect contact.

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‡Graduate Student in Mechanical Engineering, Cornell University.
[1]"The Contact-Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor," by W. H. Carrier, Transactions of the A.S.M.E., Jan., 1937.

Now from the initial assumption that the final state of the air is the same as that produced by complete mixing of the contacted and uncontacted particles, the temperature, t_2, of the air leaving a heating coil is given in terms of the initial air temperature, t_1, and the hypothetical surface temperature, t_s, by the relation

$$t_2 = Xt_1 + (1 - X)t_s \quad \ldots\ldots (3)$$

which may be solved for X to give

$$X = \frac{t_s - t_2}{t_s - t_1} \quad \ldots\ldots (4)$$

In applying Equation (4) it is assumed that the surface temperature is the same as the temperature of the steam condensing inside the coil, since the thermal resistance of the vapor film and the tube wall is much less than the air film resistance.

To evaluate the expression $\frac{t_s - t_2}{t_s - t_1}$ in terms of the basic coil properties, the heat balance for the coil must be used. In this balance the humid specific heat of air is taken as 0.245 B.t.u. per lb. deg. F and the air is assumed to be of standard density, 0.075 lbs. per cu. ft. (This last assumption is made because the heat transfer rate depends on a weight rate of flow and in the usual problem the face area of the coil and the face velocity are involved; so their product, the c.f.m., must be converted to a weight rate of flow.) Then the rate of heat transfer to the air is

$$Q = 60\ A_f\ V_s\ (0.075)\ (0.245)\ (t_2 - t_1) \quad \ldots\ldots (5)$$

and the rate of heat transfer from the steam is

$$Q = U\ A_s\ \theta_m \quad \ldots\ldots (6)$$

where A_f = the total face area of the coil, sq. ft.
V_s = velocity of air of standard density through the face area of the coil, ft. per min.
t_2 = the final air temperature, deg. F.
t_1 = the initial air temperature, deg. F.
U = the overall heat transfer coefficient from steam to air, B.t.u. per hr. sq. ft. deg. F.
A_s = the total surface area on the air side, sq. ft. = nA_{s1} for coils of Mfr. A and approximately for coils of Mfr. B.
A_{s1} = the total surface area on the air side of one row of the coil, sq. ft.
n = the number of rows of the coil in the direction of air flow.
θ_m = log. mean temperature difference, deg. F.

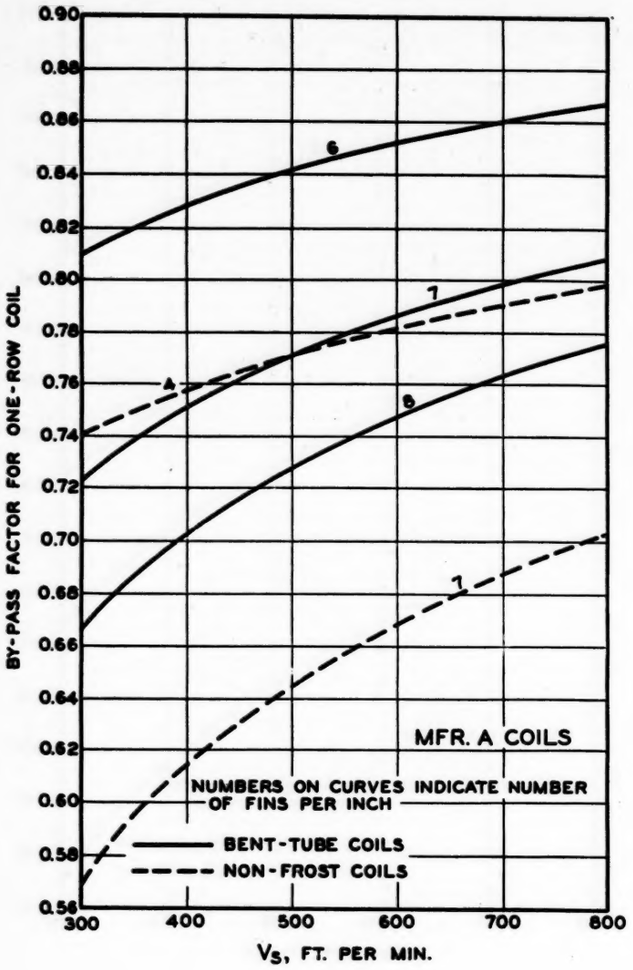
Now these two rates of heat transfer are equal; so, equating them and substituting for A_s and θ_m, we obtain

$$60\ A_f\ V_s\ (0.075)\ (0.245)\ (t_2 - t_1) = U\ n\ A_{s1}\frac{t_2 - t_1}{1n\frac{t_s - t_1}{t_s - t_2}}$$

which leads to

$$-1n\frac{t_s - t_2}{t_s - t_1} = \frac{n\left(\frac{A_{s1}}{A_f}\right)U}{1.1\ V_s}$$

$$X = \frac{t_s - t_2}{t_s - t_1} = e^{-\frac{n\left(\frac{A_{s1}}{A_f}\right)U}{1.1\ V_s}} \quad \ldots\ldots (7)$$



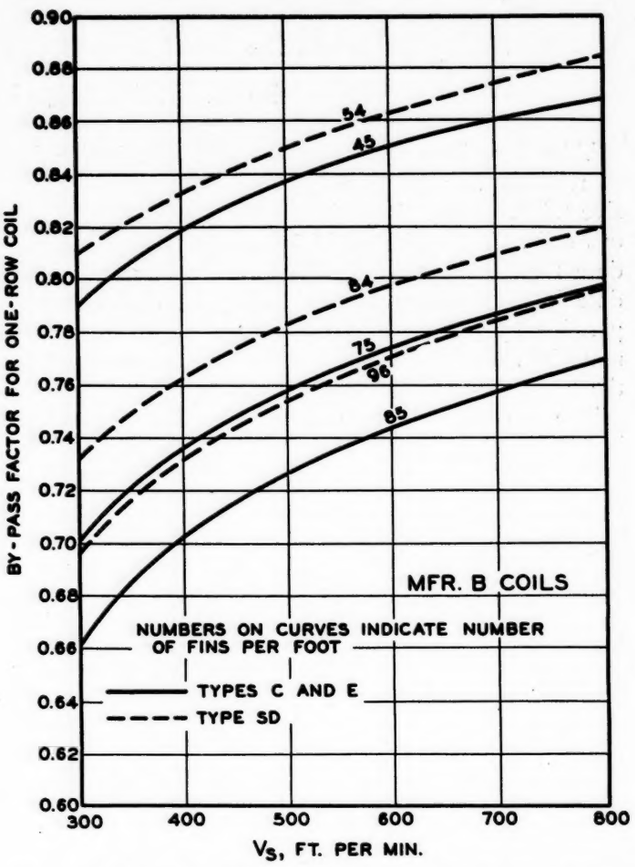


Fig. 1 (left) and Fig. 2 (above), give the by-pass factors for manufacturer A's and manufacturer B's single row coils for air of standard densities over range of economical face velocities for ordinary air conditioning systems.

The overall thermal resistance to heat transfer is composed of the resistance of the air film outside the tube, the resistance of the tube metal, and the resistance of the film of the condensing steam inside the tube. The combined resistance of the steam film and the tube metal remains substantially constant, while the air film resistance is decreased by increased air velocities. The overall heat transfer coefficient, which is the reciprocal of the overall thermal resistance, is dependent mainly on changes in air velocity for a given coil. For most coils this dependence may be expressed in the form

$$U = \frac{1}{a + \frac{b}{V_s^c}} = f(V_s) \quad \ldots\ldots\ldots\ldots\ldots (8)$$

where a, b, c are constants with c between 0.6 and 0.8.

A comparison of Equations 7 and 8 shows that the by-pass factor of a given coil is a function of V_s and n only, for $\frac{A_{s1}}{A_f}$ is approximately constant for any coil. So, for a given face velocity, the by-pass factor, X_n, for an n-row coil is related to the by-pass factor, X_1, for a one-row coil by

$$X_n = X_1^n \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The authors have prepared graphs of the by-pass factors for the coils of two manufacturers, Figs. 1 and 2. These graphs give the by-pass factors for single-row coils for air of standard density over the range of economical face velocities for ordinary air conditioning systems. Similar curves, of course, may be drawn for coils of other manufacturers. The use of these curves is best illustrated by an example.

Suppose that it is desired to heat 30,000 lbs. of air per hour from 40F to 110F using steam at 5 lbs. per sq. in. gage (temperature = 227F). For these conditions, the required by-pass factor for a single-row coil is

$$X_1 = \frac{t_s - t_2}{t_s - t_1} = \frac{227 - 110}{227 - 40} = 0.625$$

For two-row and three-row coils the required by-pass factors are respectively

$$X_2 = \sqrt[2]{0.625} = 0.790$$

$$X_3 = \sqrt[3]{0.625} = 0.855$$

Reference to the various curves of Figs. 1 and 2 shows that any of the coils listed in Table 1 may be used.

TABLE 1

Number of Rows	Type of Coil	V_s, Ft. per Min.	Required Face Area, Sq. Ft.
1	Mfr. A—Non-Frost ... 7 fins per in.	435	15.3
2	Mfr. A—Bent-Tube ... 7 fins per in.	635	10.5
	4 fins per in.	700	9.5
	Mfr. B—Type C or E. 45 fins per ft.	305	21.9
	Mfr. B—Type C or E. 75 fins per ft.	725	9.2
	Mfr. B—Type SD ....84 fins per ft.	545	12.2
	Mfr. B—Type SD ....96 fins per ft.	745	9.0
3	Mfr. A—Bent-Tube .. 6 fins per in.	645	10.3
	Mfr. B—Type C or E. 45 fins per ft.	650	10.3
	Mfr. B—Type SD ...54 fins per ft.	535	12.5

Now the required face area of the coil may be found from the required face velocity for air of standard density and the known weight of air to be heated. If M represents the number of pounds of air to be heated per hour, then

$$M = (60)\ (0.075) V_s\ A_f$$

$$A_f = \frac{M}{4.5\ V_s}$$

This gives the actual face area required. The actual face velocity, V_a, is not equal to the face velocity for air of standard density, but is related to it by

$$V_a = V_s \frac{\rho_s}{\rho_a}$$

where ρ_s = density of standard air, lbs. per cu. ft.
= 0.075 lbs. per cu. ft.
ρ_a = density of air under actual conditions, lbs. per cu. ft.

The final choice of the coil for the installation of the problem would depend on the initial cost of the coil and on the operating cost of the coil, i.e. the cost of moving the air past the coil. For determining the cost of moving the air, the friction drop in pressure through the coils must be known. These pressure drops are given for the various coils in Figs. 3, 4, 5, and 6. These curves are for air of standard density and the actual friction drop must be determined from them by use of the following relation:

$$\Delta p_a = \Delta p_s \left(\frac{\rho_s}{\rho_a}\right)$$

When the pressure drop, the cost of energy, and the number of hours of operation are known as well as the cost of the coils, the selection of the most economical coil may be made.

The equivalent by-pass principle is also useful in predicting the performance of coils of a given design. By testing one coil at several different air velocities with steam at one temperature, only, and for one state of the entering air, only, the equivalent by-pass factor of a one-row coil may be found as a function of the air velocity through the face area. Then the state of the exit air may be predicted for any face velocity, number of rows in series, steam temperature, and state of the entering air.

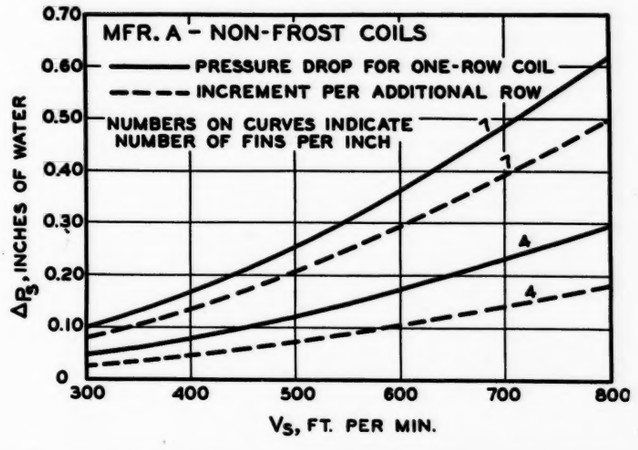
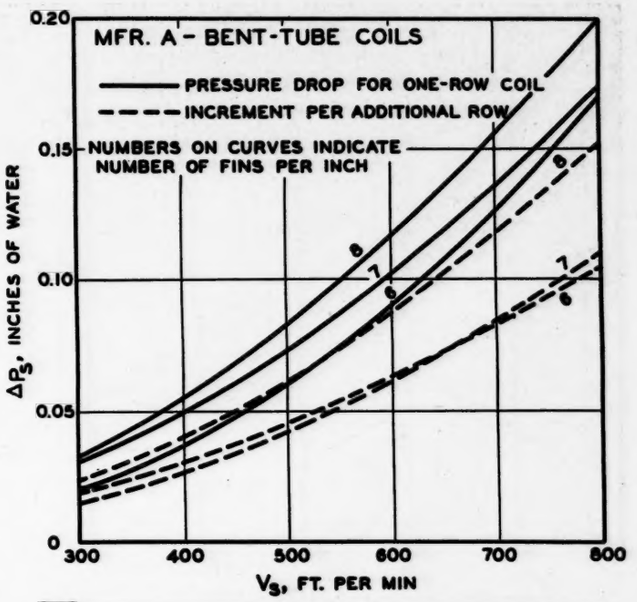
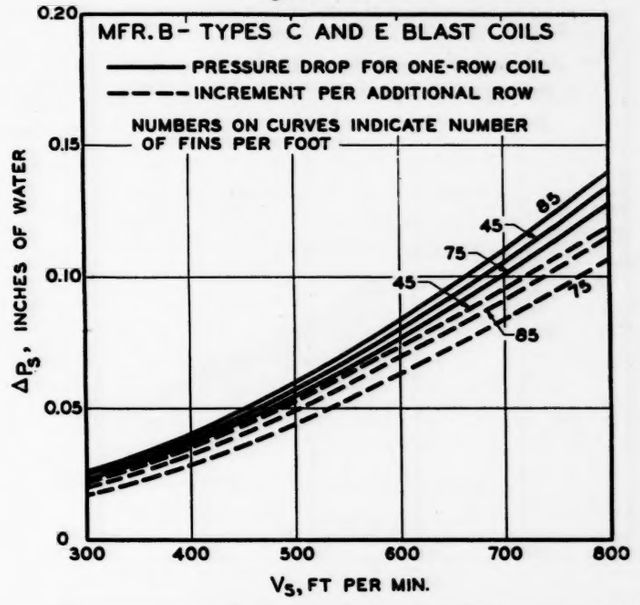
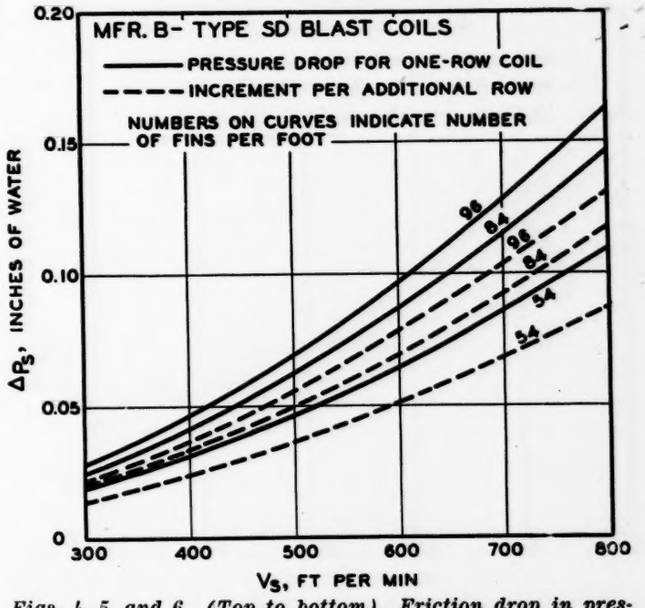


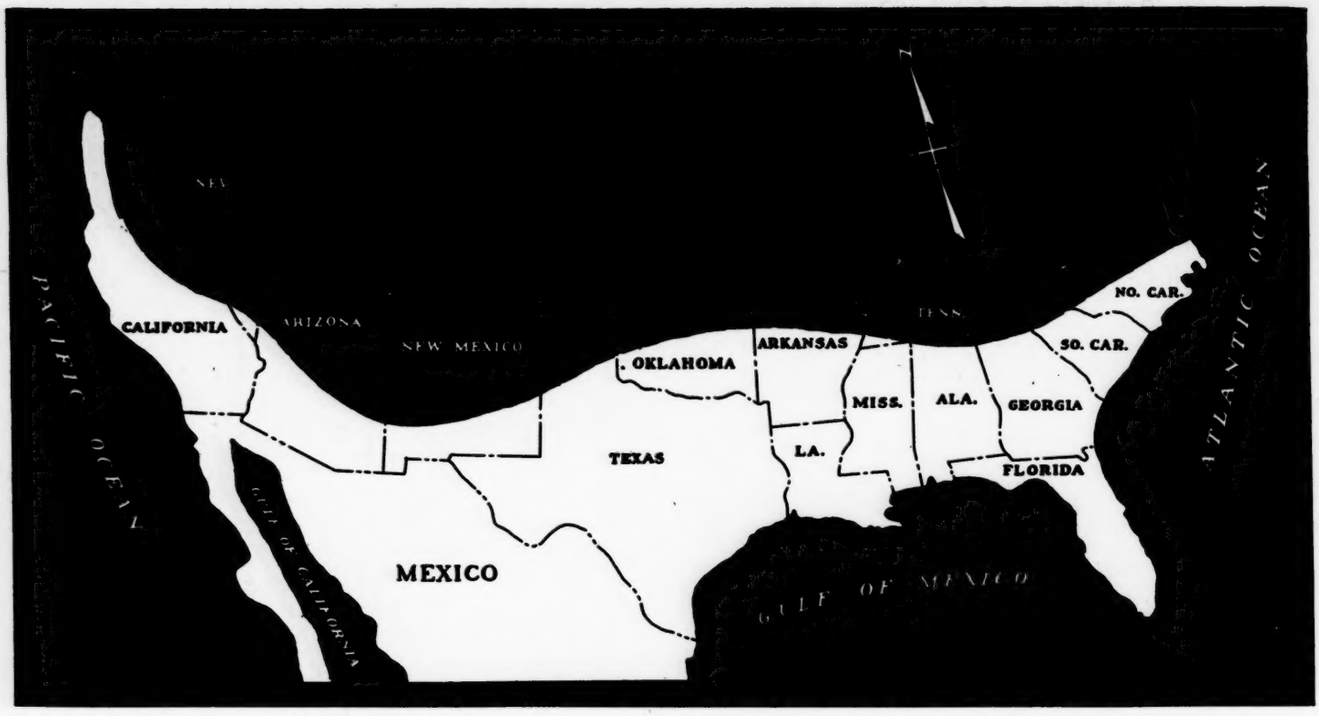
Fig. 3. Friction drop in pressure through manufacturer A's non-frost coils.







Figs. 4, 5, and 6. (Top to bottom). Friction drop in pressure through manufacturer (Fig. 4) A's bent-tube coils; (Fig. 5) B's types C and E blast coils; and (Fig. 6) B's type SD blast coils.



White areas show regions in thirteen Southern States where solar water heaters are successfully used.

Solar Heaters for Service Water

By ANDRE MERLE†

THE sun is our most important source of energy, and life would not be possible without it. Scientists and engineers have been trying for many years to harness but a small fraction of its energy. Among these attempts are those made to produce hot water and thus avoid using fuel for the purpose.

While the use of solar water heaters is by no means new since there are records of these heaters having been used regularly for the past 40 or 50 years in this country, still there has been a considerable revival of interest in these heaters recently. For example, a three-year survey and study by the writer indicates that there are close to 25,000 solar water heaters in use in the South at present. Some estimates have run as high as 25,000 in Miami, Florida, alone. There are said to be in the city of Miami no less than ten organizations prepared to undertake installations of sun heaters. Thus, it is quite an industry in some parts of the Southern United States.

It is also reported that the federal government has installations in some of its low-rent projects in Florida. Also, the United States War Department is said to have large installations both in the continental United States and in the island possessions.

The object of this article is to present some of the interesting data regarding present day solar heaters including types and forms in common use; some information about the materials commonly employed in their construction and about details such as heater location and arrangement of the several parts together with some information regarding the dimensions and capacities found workable.

In considering the possibility of making use of the sun for warming hot water it is important to know something about the nature and amount of heat.

Experience indicates that the heating ability of the sun increases with the length of day and with its position in the sky at noon. We know also that the heating effect is less intense during the winter season than during the summer. Yet, the sun is closer to the earth during the winter than it is during the summer (see Fig. 1). The angle at which the rays strike the earth is what regulates heating ability. Vertical rays are more intense than angular rays.

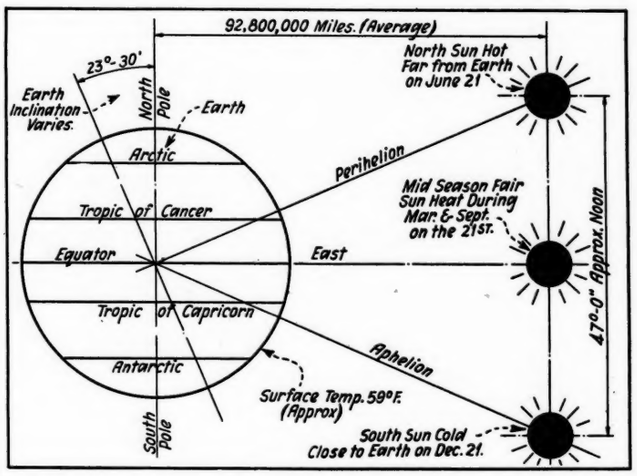


Fig. 1. Sun's position relative to the earth in each season.

†Consultant on Mechanical Equipment, Washington, D. C.

Another point which should be kept in mind is that the sun heat is available for heating water only during the daylight hours. Heat from this source is not available at night nor on dark, cloudy or stormy days.

As to the amount of sun received on the earth's surface, a figure frequently used is 7.15 B.t.u. per sq. ft. per minute at sea level when the surface receiving the heat is perpendicular to the sun's rays. It will be seen that this corresponds to 430 B.t.u. per sq. ft. per hour. Data covering the amount of heat actually received from the sun are still not in good shape. There seems to be no information available regarding the amount of sun energy received during the summer as compared to winter.

These several considerations indicate that solar heaters must meet the following requirements:

(1) The heat receiving surface must be placed at the best possible angle to receive a maximum of the sun's heat.

(2) Materials should be selected so that they will absorb as much of the sun's heat as possible.

(3) Since heat can be received only during the few hours of the day and these hours do not coincide with the hours when water is required, some form of storage must be provided.

(4) Heated water must flow from the heat absorbing surface by gravity circulation in most cases and consequently the heat absorbing surface must be placed lower than the storage tank or collecting device.

We can now turn attention to some of these points considering them one at a time.

Solar heaters can be classified according to the method of installation, as follows:

A—Flush type on roof, (New buildings) (Fig. 2)

B—Still type on roof, (Old buildings)

C—Awning type, extended from side over windows, doors, (Building or garage) (Fig. 3)

D—Flat roof or ground type, (Fig. 4).

The installation may use a vertical or horizontal hot water storage tank, within building proper or fully exposed, or semi-exposed, such as is possible with a false chimney arrangement.

Regardless of the type of heater used, opaque glass is commonly selected to cover the heat absorbing surface. This glass is usually either straight or curved and is the so-called single strength glass. It should be kept in mind that glass does not prevent the passage of heat outward from the system during the night and consequently there is usually a small loss of some of the stored heat during the hours when the sun is not shining. Opaque glass is considered a good material since a large part of the sun's heat energy is trapped under it (see Fig. 5). Estimates place the amount of heat passing through opaque glass at about 300 B.t.u. per sq. ft. per hour or about 2000 B.t.u. per day when the glass is in a plane perpendicular to the sun's rays at noon in mid-summer at about 27° North latitude. Additionally, for year-round use in the 27° North latitude zone it is generally considered safer to assume that 1000 B.t.u. per day will pass through each square foot of the exposed glass surface. As mentioned, the greater part of this admitted heat is trapped under the glass although small amounts are lost.

Fig. 2. Flush type solar heater on tile roof. Note false chimney which holds a vertical storage tank. Tank should be set at least 2 1/2 ft. above solar heater.

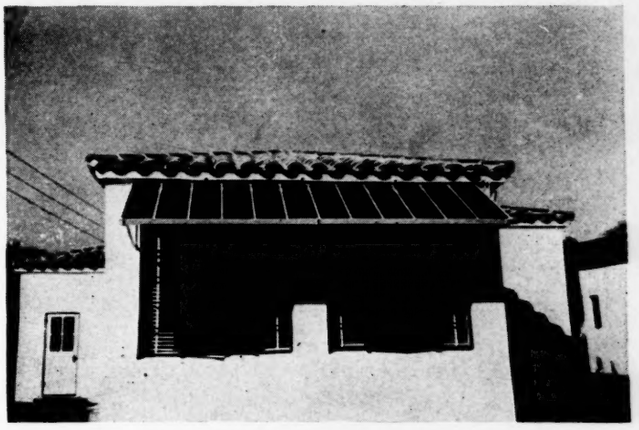


Fig. 3. An awning type solar heater which serves a double purpose—to shade the window and to heat the service water. An insulated storage tank beneath the roof stores the water until needed.

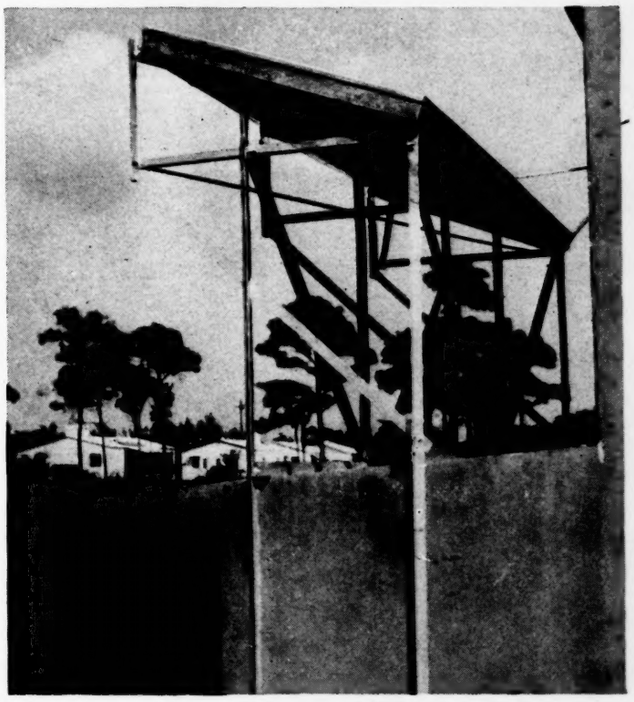


Fig. 4. Solar heater located over storage room in back of main roof due to fact front side of roof is shaded. Heater is raised so as to absorb sun's rays over rear section of roof.

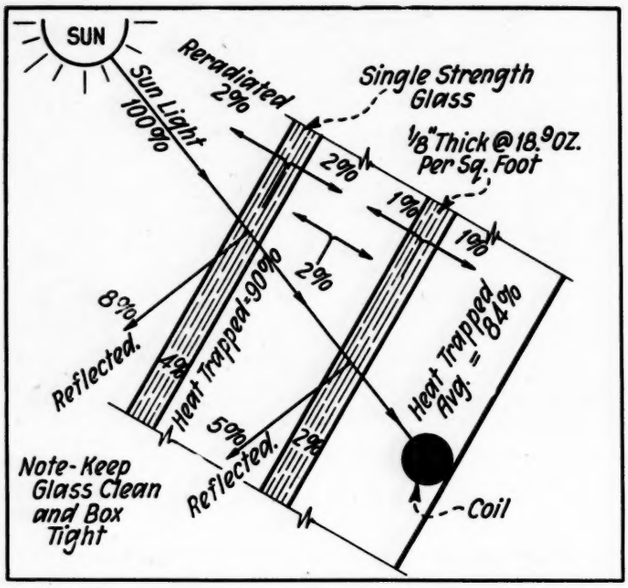


Fig. 5. Diagram showing typical heat flow through glass. In the Miami area approximately 1000 B.t.u. per day is estimated to pass through each square foot of exposed glass area. About 84 per cent of this heat is trapped in the coils.

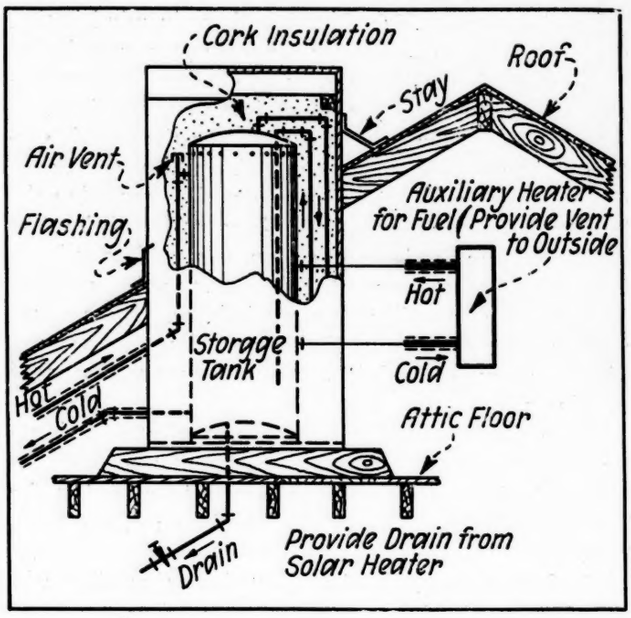


Fig. 7. Diagram showing method of locating the storage tank in a false chimney with tank projecting through roof to get height necessary to permit gravity circulation.

As regards materials for other parts of these heaters, it may be said that a typical solar water heater includes a flat, air and water-tight metal box, heat insulated, and glass covered. In appearance this box resembles those sometimes used as hot beds for growing plants (see Fig. 6). The inside of this flat box is usually copper, painted a flat black. Pipe or tubing is laid on the copper surface and is sometimes brazed or spot-welded to the copper. Pipes or tubes are also painted a flat black. The pipes are carefully fabricated with return bends. The whole is so fabricated that the water circulates by thermo-syphon although large solar heaters sometimes use power driven circulating pumps instead of depending on the thermal action of the heated water.

As a general rule the cold water enters at the low end of the pipe coil, moves upward as heated, and leaves the coil at the top, flowing from there to a storage tank which in the case of gravity circulation should be set about 2 ft. 6 in. higher than the top or outlet connection of the pipe coil (see Fig. 7).

The storage tank should hold a full day's demand based on not less than 15 gallons per person per day.

While the majority of solar water heaters in use today are of the single sash, straight pipe, thermo-syphon construction, large installations are frequently double circuited with a circulating pump. It might be remarked in passing that large installations are said to be tricky to design and to balance and that a considerable element of experience is necessary in order to make them function properly.

Generally, the pipe or tubing used is plain round pipe although a plain oval shaped tubing is employed as are also finned type tubes in some cases.

It has already been mentioned that the angle at which the absorbing device is placed is an important matter and it can also be readily seen that it is very important that the absorbing part of this heater shall be so located that it will not be shaded.

As regards the angle of inclination of this absorbing box in relation to the sun's rays, it should be kept in mind that the problem of absorbing enough heat is

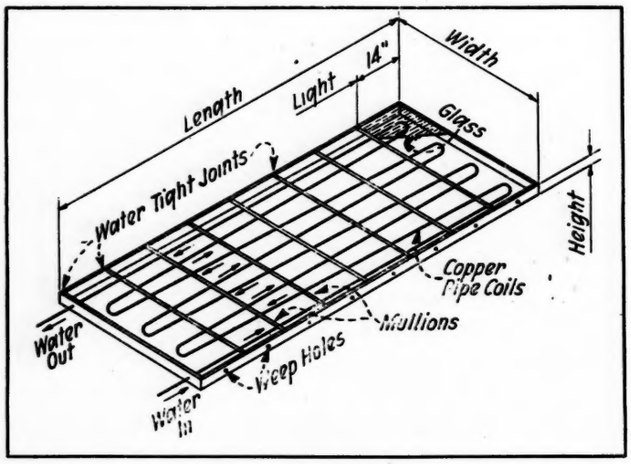


Fig. 6. Detail of principal parts of an 8-light heater section. Box is insulated, made water tight and glass covered.

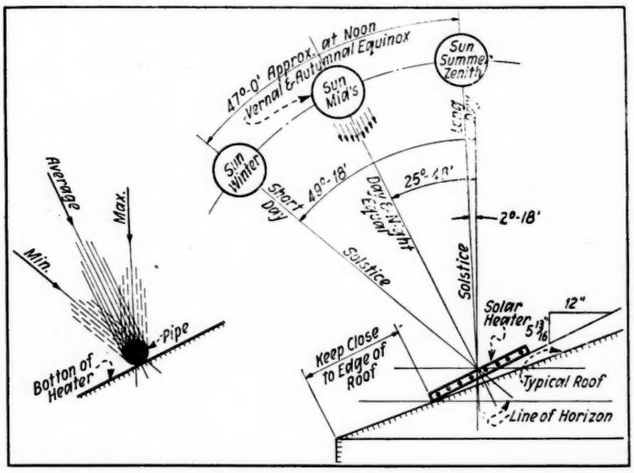
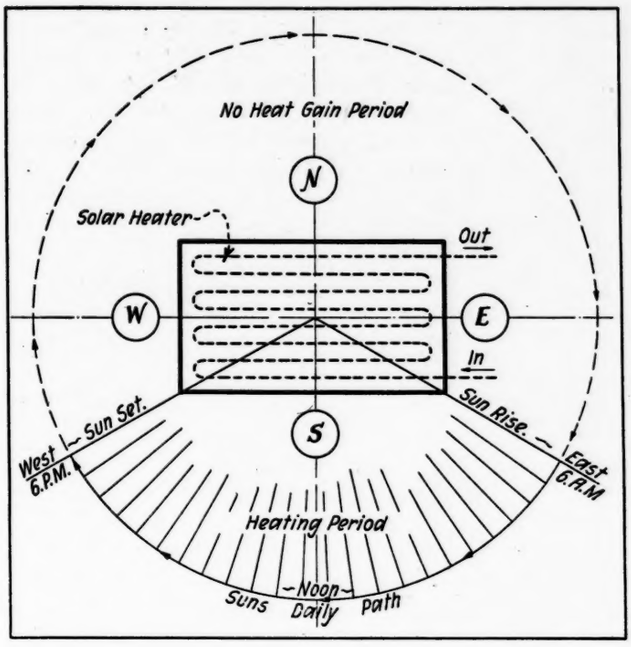


Fig. 8. Angle of incidence for maximum yearly efficiency of heater in Miami, N. Lat. 25° 48' and 25 ft. above sea level.



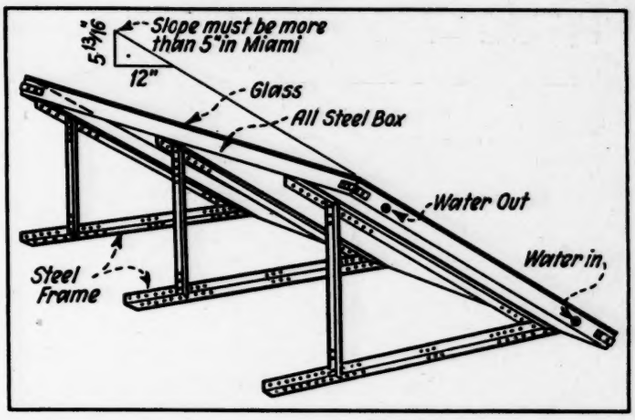


Fig. 9. (left) Sketch showing importance of locating heater to face south with the water tubes running east and west.

Fig. 10. (above) Typical installation on flat roof or yard.

more important during the winter than during the summer.

Figs. 8 and 9 indicate some of the geometry of the angle at which the absorbing box can best be located. In Fig. 8 is shown the proper angle of inclination for the vicinity of Miami, Florida, and it should be noted that the vernal-autumnal angle was selected as a sort of compromise between the ideal angle for either summer or winter conditions. Fig. 10 indicates how the selected angle of inclination is applied to a heater located on a flat roof or yard. An arrangement for varying the angle at different times of the year would no doubt be valuable but considerations of cost and complexity of mechanism usually rule out anything more than whatever adjustments can be simply made. In the great majority of installations the angle is fixed and no adjustment is possible.

The following hints regarding location of the heat absorbing box are worth attention. In addition to selecting a sunny location the most popular place to put these boxes is on a sloped roof as high up as possible and still keep them low enough so that the storage tank can be installed above them. Ground installations are usually not recommended as they not only permit tampering but they may eventually become shaded by trees. The heat absorbing box should face South but where this is not possible one unit may be installed facing East and another installed facing West. Generally large heat absorbing boxes are required when they are installed facing otherwise than South.

A good idea of some of the locations selected for the boxes can be obtained from the accompanying photographs. In Fig. 2 a false chimney can be seen. Such a chimney is built around the storage tank which may be placed vertically inside it. Where storage tanks are placed horizontally they can frequently be put under the roof surface and thus avoid the necessity of constructing a false chimney. The roof loading should be examined to be sure that the heat absorbing box and the storage tank do not overstrain the structure.

Since it is so difficult to establish accurately the amount of heat which can be made available to a given heater, it follows that the design is more or less empirical. It is believed however that use of the design data in the accompanying tables will result in a full tank of heated water within six hours of average sunshine and with a minimum temperature rise of 30F within the zones outlined in the map at the beginning of this article. It is important that the design and construction data given in the accompanying tables be followed as closely as possible. An oversized absorbing device will produce excessive heat while undersizing

TABLE 1—HOT WATER USAGE
(Approx. 15 gallons per person per day)

Residences			Apartments			
Number of		Capacity, Storage Tank, (Gal.)	No. of Apts.	Total Number of		Capacity, Storage Tank, (Gal.)
Bed Rooms	Bath Rooms			Bed Rooms	Bath Rooms	
1	1	66	4	4	3	250
2	1	82	6	6	4	400
3	1	100	8	8	6	500
4	2	120	12	12	8	2-400
5	2	150	16	16	12	2-500
6	3	200	20	20	15	3-400
8	4	250	30	30	22	3-500
10	4	300	36	36	27	4-500

TABLE 2—STANDARD SOLAR HEATER DIMENSIONS

No. of Lights	Lineal Feet of Tubing	Overall Length of Heater Box	Capacity, Storage Tank (Gal.)
7	68'-4"	8'-7½"	52
8	77'-8"	9'-10"	66
9	87'-4"	11'-0½"	74
10	97'-0"	12'-3"	82
11	106'-8"	13'-5"	92
12	116'-4"	14'-8"	100
13	126'-0"	15'-11"	120

Each light (Glass) is approx. 14" wide by 48" long. Boxes are about 3¼" high and 48" wide. Heaters weigh about 50 lb. per sq. ft. on roof.

TABLE 3—DIMENSIONS AND WEIGHTS OF HOT WATER STORAGE TANKS

Capacity, Tank (Gal.)	Tank Dimensions		Dimension, Tank with Insulation		Weight of Tank and Insulation (Lb.)
	Dia., In.	Height	Width x Depth	Height	
66	18	5'-0"	2'-4"x2'-4"	5'-10"	800
82	20	5'-0"	2'-6"x2'-6"	5'-10"	1050
100	22	5'-0"	2'-8"x2'-8"	5'-10"	1300
120	24	5'-0"	2'-10"x2'-10"	5'-10"	1600
150	30	4'-0"	3'-4"x3'-4"	4'-10"	2000
200	30	5'-4"	3'-4"x3'-4"	6'-2"	2600
250	36	4'-10"	3'-10"x3'-10"	5'-8"	3150
300	42	4'-3"	4'-4"x4'-4"	5'-1"	3700
400	42	6'-0"	4'-4"x4'-4"	6'-10"	4900
500	42	7'-0"	4'-4"x4'-4"	7'-10"	6050

TABLE 4—NUMBER OF STANDARD HEATERS REQUIRED TO PROVIDE HOT WATER

Gal. Stored	Facing South			Facing E. and W.			No. of Tanks
	No. of Lights per Section	No. of Heater Sections	Approx. Overall Length	No. of Lights per Section	No. of Heater Sections	Approx. Overall Length	
66	8	1	9'-9"	7	2	17'-3"	1-66
82	10	1	12'-2"	8	2	19'-8"	1-82
100	12	1	14'-7"	10	2	24'-6"	1-100
120	8	2	19'-6"	12	2	29'-4"	1-120
150	10	2	24'-4"	13	2	31'-10"	1-150
200	12	2	29'-2"	10	4	49'-0"	1-200
250	10	3	36'-6"	11	4	53'-8"	1-250
300	12	3	43'-9"	12	4	58'-8"	1-300
400	12	4	58'-4"	12	6	88'-0"	1-400
500	12	5	72'-11"	11	8	107'-4"	1-500
800	12	8	116'-8"	12	12	176'-4"	2-400
1000	12	10	145'-10"	11	16	214'-8"	2-500
1200	12	12	175'-0"	12	18	264'-0"	3-400
1500	12	15	218'-9"	12	20	293'-4"	3-500
2000	12	20	291'-8"	12	30	440'-0"	4-500

results in little if any hot water. Examination of these tables will indicate that there is a reasonably constant ratio of square feet of glass to linear feet of copper tubing to gallons of water heated and stored in the storage tank. Care must be exercised to establish the proper relation between the angle of inclination and the gravity flow. The force producing flow is extremely small and a delicate balance must be arrived at to secure satisfactory economical operation.

It should also be mentioned that with the design data included with this article a succession of two or three days without sunlight would result in the absence of warmed water. Where there are that many successive days without sunlight an auxiliary method of heating is definitely indicated.

If a solar heater is to operate satisfactorily some attention must be paid to it in order to see that it is kept in good condition. One of the most important points is to see that the opaque glass covering the heat absorbing box is kept clean. A short piece of rubber hose and a convenient water connection will simplify this task very greatly and will increase the water temperature. A dusty, dirty sash also retards quick heating up.

The box should be checked for rain and dust leaks occasionally as leaks reduce heat absorbing efficiency.

Scientists Find Six Periodic Fluctuations in Sun Constant

At least six periodic fluctuations in the so-called solar constant, all of which are included in a ten-year period, have just been reported to the National Academy of Sciences by a group of Harvard University mathematicians after an intensive analysis of observations made over an interval of 15 years by Smithsonian Institution observers.

The solar constant is the amount of solar radiation —heat, light, and ultra-violet radiation—which would be absorbed each minute by a black cube one cubic centimeter in size placed at the outer edge of the earth's atmosphere. It is essentially the amount of solar radiation which would fall on a square centimeter of the earth's surface if the planet, like its satellite the moon, had no atmosphere.

This solar constant is approximately 1.940 calories. If the heat output of the sun was constant, this figure would never fluctuate. For several decades, however, there has been accumulating evidence gathered by Smithsonian observers that it does fluctuate, that these fluctuations are rhythmic, and that consequently the sun must be a variable star.

The Harvard investigations were made entirely without reference to any of the previous findings. Altogether they found ten solar variation cycles. Six of these upon which chief emphasis is placed are almost exact submultiples of 10.2 years. This is highly suggestive of the sun-spot cycle, during which the number of titanic cyclones in the atmosphere of the sun goes from a minimum to a maximum and back again.

The length of the sun-spot cycle is generally considered to be approximately 11.3 years. This is close to the average since observations were started. The period, however, varies in length, and it so happens that the two sun-spot periods, parts of which fell within the 15-year sample period for which the records were analyzed, averaged 10.1 years. Thus the six fluctuations would seem to be components of the major sun-spot cycle itself—due fundamentally to the same unknown causes.

ESTIMATING HEAT FLOW THROUGH FLAT ROOFS

FLAT ROOFS

	COLOR OF ROOFING	CONSTRUCTION NO. (ON H. & V.'S REFERENCE DATA SHEET NO. 77 & 79)	THICKNESS OF ROOF DECK	INSULATION	PLASTER CEILING	HOURS OF DAY: 8 AM	10	12 NOON	2 PM	4	6	8	10	12
		WHEN TEMPERATURE DIFFERENCE GIVEN BELOW IS MULTIPLIED BY THE OVERALL COEFFICIENT, U, THE PRODUCT IS THE ACTUAL RATE OF HEAT FLOW TO INDOOR AIR, IN B.T.U. PER HR. SQ. FT.											
CONCRETE ROOFS	BLACK	100 A	2"	0	NO	39	63	76	74	62	47	33	20	10
		101 A	4"	0	NO	23	46	64	72	65	52	39	27	17
		102 A	6"	0	NO	15	32	52	64	63	54	43	32	22
		101 E	4"	2"	NO	14	29	48	61	62	55	44	33	24
		102 E	6"	2"	NO	12	25	43	57	60	56	45	35	25
		112 E	4"	2"	YES	13	28	46	59	62	55	44	34	24
		113 E	6"	2"	YES	12	24	41	55	60	55	46	35	26
	ALUMINUM	100 A	2"	0	NO	16	27	34	35	31	25	19	12	7
		101 A	4"	0	NO	11	21	29	35	33	28	23	18	12
		102 A	6"	0	NO	7	13	22	28	29	27	23	18	13
		101 E	4"	2"	NO	6	12	20	27	29	27	23	19	14
		102 E	6"	2"	YES	6	11	18	25	28	27	23	19	15
		112 E	4"	2"	YES	6	12	20	26	28	27	23	19	14
		113 E	6"	2"	YES	6	10	18	24	27	27	23	19	15
WOOD ROOFS	BLACK	104 A	1½"	0	NO	23	45	63	70	64	52	39	27	17
		104 E	1½"	2"	NO	11	22	39	53	59	55	46	36	27
		115 E	1½"	0	YES	11	21	38	53	58	55	46	37	25
	ALUMINUM	104 A	1½"	0	NO	9	19	27	31	31	27	21	16	10
		104 E	1½"	2"	NO	6	9	16	23	27	26	24	20	16
		115 E	1½"	2"	YES	6	9	16	23	26	26	24	20	15

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H. & V.'s REFERENCE DATA—179

ESTIMATING HEAT FLOW THROUGH SUNLIT FLAT ROOFS

The table on the other side of this sheet, prepared by C. O. Mackey and L. T. Wright, Jr., based on a method outlined in their series of articles in HEATING & VENTILATING can be applied for finding the amount of heat flow through certain flat roof constructions at various hours of the day. The table is based on a value of 4 B.t.u. per hr. sq. ft. deg. F for the outside air film coefficient and consequently, when used with the overall coefficients given in HEATING & VENTILATING's Reference Data 79, the calculated heating loads are on the safe side.

The table is calculated for a North latitude of 40 degrees but it is believed that the effect of latitude for any city in the United States is slight and probably less than 3%. The table is also calculated on an indoor D.B. temperature of 80F. If another indoor dry bulb air temperature, t, is used, add (80 — t) to the temperature differentials given in the table on the other side.

It should also be noted that the table is calculated for a typical design day in mid-July in which the outside D.B. temperatures vary from hour to hour, as shown in Fig. 1 below.

It is believed that the table with its rational basis should give the combined contribution of air temperature difference and sun effect to the cooling load at any hour of the design day.

Similar tables for certain walls of one material will be found in HEATING & VENTILATING's Reference Data 175-176, and for certain wood frame walls, in HEATING & VENTILATING's Reference Data 177-178, in which a method of estimating the sun effect on roofs is presented.

Attention is also called to HEATING & VENTILATING's Reference Data No. 83-84 in which a method of estimating the sun effect on roofs is presented. The data on this sheet, to some extent, supersedes the information given there.

Example: What is the heat load at 4 p.m. and 6 p.m. on a typical sunny July day due to sun heat and air temperature difference on the flat concrete roof of a building, constructed with a roof deck 4 in. thick, having 2 in. of insulation and a plastered ceiling? The surface of the concrete is covered with an aluminum colored paint. The roof area exposed to the sun is 20,000 sq. ft.

SOLUTION: It will be seen that the roof construction designated is 112E in HEATING & VENTILATING's Reference Data 79-80, and the value of the coefficient U is 0.12 B.t.u. per hr. per sq. ft. The rate of heat flow to the indoor air at 4 p.m. can be calculated as follows:

At 4 P.M.

$$20{,}000 \times 0.12 \times 28 = 67{,}200 \text{ B.t.u. per hour.}$$

Similarly, the rate of heat flow to the indoor air at 6 p.m. can be calculated as follows:

At 6 P.M.

$$20{,}000 \times 0.12 \times 27 = 64{,}800 \text{ B.t.u. per hour.}$$

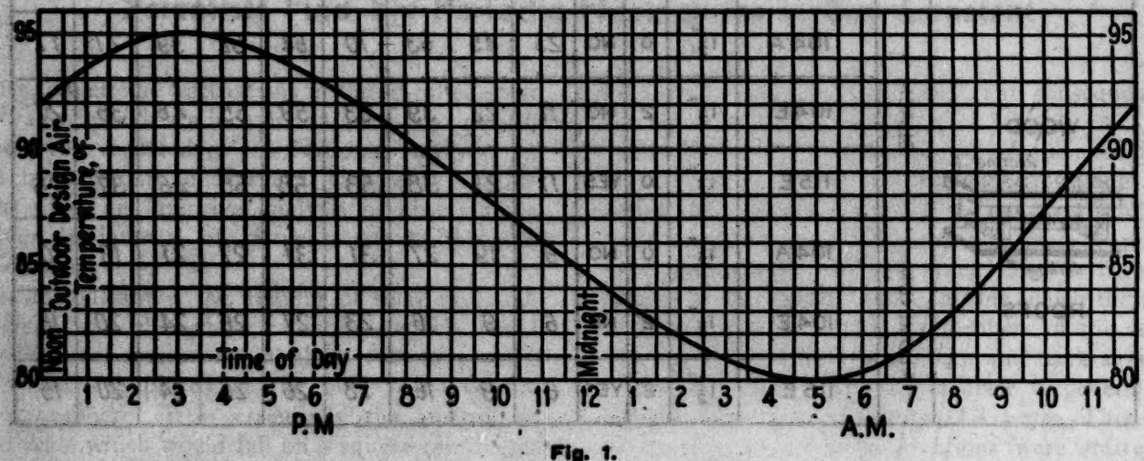

Fig. 1.

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Facts and Fallacies in Heating Piping

PART 5—HEADERS AT THE BOILER

By T. W. REYNOLDS

IT is much easier to obtain an agreement among engineers as to the best method of connecting the return end of a boiler than it is in the case of the steam end. There are at least eight points of discussion involved and these may be outlined as follows:

Is it better to drop the boiler leads into a header, Fig. 15(a), than it is to connect them straight into the side of the header, Fig. 15(b). Should the main rise from header and then drop again before proceeding, Fig. 15(a), or just rise off header, Fig. 15(b), or again is it all right to omit both rise and drop and take the main straight from the side of the header, Fig. 15(c)?

Is it necessary to provide for expansion with swing joints, Fig. 15(a), or Fig. 15(d), or are straight connections from boiler to header and from header to main good enough, Fig. 15(c); or can we go still further in this respect and install the header directly over the boiler outlets, Fig. 16(a), so as to save due to reduction in number of large fittings with their covering and labor?

Will the job work if the header is omitted entirely and each main is connected separately to a boiler outlet, Fig. 16(b)? Finally, should the main be taken off the header so as to cause both steam and water to flow in the same direction with the header, Fig. 15(a), or will it work just as well to connect the main from one end of the header and the drip from the other end, Fig. 16(c), in spite of the fact that the steam and water will then be counterflow?

First it may be said that it is customary for various published works to illustrate Figs. 15(a), 16(c), and 15(d), as standard practice with preference in the order named, but in actual practice we find installations as shown in all sketches, with variations one from the other and no two jobs exactly alike when viewed in succession. Most of the smaller installations inspected will be found as in Fig. 16(b), yet in most cases giving good enough results, with many others as in Fig. 16(a) and all working satisfactorily. At the same time in many large installations boilers will be found connected as shown in Fig. 15(b) with very good results.

It is interesting to observe when anyone is wont to criticize the piping in Fig. 16(a) as too rigid, that many boilers are of the header type as in Fig. 16(d) and therefore must be assembled with a return header on each side and a common steam header on top with as many rigid connections to each header as there are sections in the boiler, and this rigidity occurs at the hottest point in the entire system.

In some cases where a boiler is without a header and there are two mains connected separately and direct to the boiler outlets, Fig. 16(b), one may find one of these mains to be the most favored in steam circulation. This is more likely to happen on an automatic installation and in such cases it will usually be found that this main is most favored because it is connected to the hottest section, such as an uptake section or a

A NOTE BY THE AUTHOR

Standard piping connections are slavishly reproduced and passed on from one author to another. Quite often they also form an impressive part of our plans and a sacred niche in the specifications. Here they receive most honorable mention before being passed on to the contractor, yet the advantage of many of these connections seems extremely doubtful. This, we do know: they serve but to increase the cost.

These so-called standard piping connections were not originated in this era, but are hangovers from opinionated ideas of long ago which having gained vogue are now hard to dispel. They show little or no change from those thirty years back, notwithstanding the rapid changes that are continuously taking place in practically all other phases of heating.

The need for each of these connections is commonly defended by certain assumptions, or by a line of reasoning which off-hand seems logical because the reasoning is not pursued far enough. Such reasoning might have had some justification in other days where water in pipes was a troublesome factor and when but little was definitely known of what actually took place inside a pipe or fitting. Today we have intensive research with glass pipes, colored liquids and with pipes whose arrangement may be quickly changed at will so as to produce any desired condition for experiment.

Experience acquired in contracting, as well as in trouble shooting over a long period of years at the rate of several trouble jobs per day, eventually caused the writer to show no hesitation in repeatedly violating with impunity many of these sacred traditions with no harmful results. With this in mind, the elimination of waste, and in some cases betterment of method, it is the writer's thought to present herein a number of changes in piping methods which the writer has found along with some others to be the better way. In general these piping methods refer to connections as used in a one pipe steam heating system. Connections common to other systems are so noted, or if not their application is obvious.

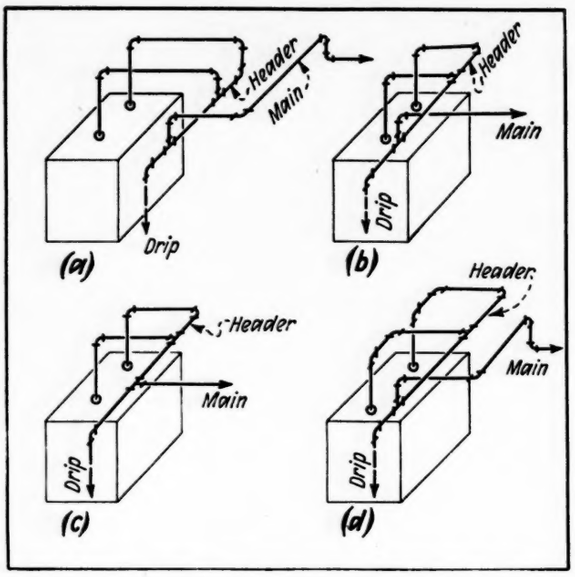


Fig. 15. Boiler headers which give satisfactory results. Figs. 15 (a) and 15 (d) represent standard practice while Figs. 15 (b) and 15 (c) represent common practice and greater simplification.

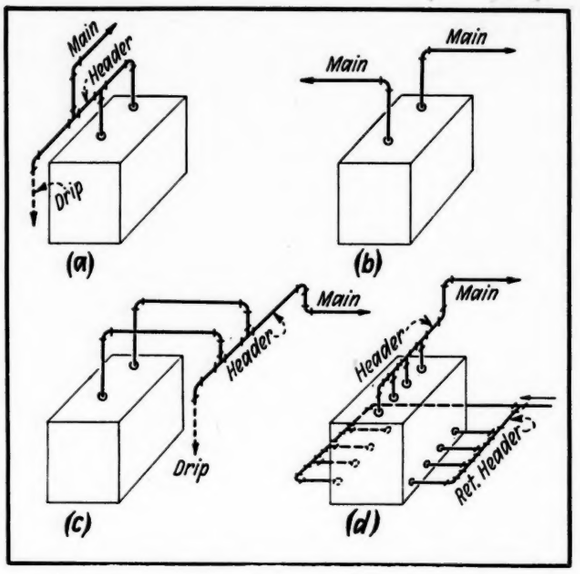


Fig. 16. Various forms of boiler headers. Figs. 16 (c) and 16 (d) represent standard practice while Figs. 16 (a) and 16 (b) represent occasional practice and greater simplification.

rear section which has much more heating surface.

Upon occasion this weakness of a headerless boiler may become a strong point and be turned to good advantage simply by altering an overloaded main so as to connect with this hottest section. In new installations the longest main or the one with the greatest load can be similarly connected. Generally speaking, however, the writer's judgment based on extended experience in alteration work is that under proper conditions the effect of the hottest section becomes too slight to unbalance heating conditions. By proper conditions is meant clean water and correct pipe size of main with rise to main full size of boiler outlet.

It has been said that a steam header is required in order to insure that the boiler functions as a whole, since it is argued that if only a single outlet were used the section to which it is connected would do most of the work. Maybe so to some slight extent in a laboratory, but on the other hand we are continuously revising our opinions with lapse of time just as many engineers now feel that a header is no longer necessary in order to provide a reservoir of steam for a sudden demand. This demand occurs only during the heating up period, at which time a lot of header and boiler pipe and fittings of large size and much greater weight add to the pick-up load and therefore to the demand for steam. In many cases this tax may be all out of proportion to the piping elsewhere in the system where pipes are more lengthy, but of much smaller size.

There may be some excuse for storage of steam in high pressure industrial systems where large instantaneous demands are customary and where a small amount of steam with high energy is compressed in a relatively small space, due to its great pressure, but in a low pressure heating system the storage of just one lone pound of steam would require a header of 10 in. diameter and 13 feet long. In best practice, headers are no longer sized for storage, but rather on the same basis as the general run of piping in the system, that is, in accordance with the maximum load carried by any portion of the header. A large header is also thought necessary in order to handle any entrained water which might be carried over from the boiler. The answer to this is contained in the next paragraph in the discussion of another detail of boiler piping.

There seems to be no advantage in dropping the leads into the header, Fig. 15(a), instead of connecting them into the side as indicated in Fig. 15(b), provided the vertical leads are sufficient in number and full size of boiler outlets so as to keep the velocity of steam within these verticals within proper limits, thus allowing the entrained water to fall back to the boiler. Where headroom is limited the method of piping shown in Fig. 15(a) may put the header down too close to boiler water line and if the main is also dropped as shown in Fig. 16(c) it is fairly certain that before it terminates the main will be closer to the water line than desirable.

In such a case it may be necessary either to put the boiler in a pit at considerable expense if rock or water is encountered, or else the pipes will have to be of larger size so as to decrease the pressure drop and therefore decrease the required distance above the water line. Many jobs work satisfactorily with piping 18 in. above the water line and occasionally one may be observed with this distance reduced to only 12 in., yet giving no trouble. It's all in the matter of ample sizing so as to keep down the pressure drop, just as a main near the boiler may be dripped only a little above the boiler water line, because there is no appreciable pressure drop up to this point.

As for keeping the steam and water flow in parallel within the header, Fig. 15(a), the writer feels that where outlet verticals are made full size of boiler outlets as previously mentioned, there is no more necessity for this than there is at any other point within a one pipe steam heating system. By the same token the main can be taken from the side of the header instead of from the top, especially if there are two mains, for

neither of these would be the same size of the header. Consequently there would be less liability of these mains taking water from the header. The header is graded to the drip and if thought necessary the main can be graded up to its first connection, after which it can be graded down to its end in accordance with customary practice. This may save a boiler pit.

Cast iron boilers usually have rather large return inlets, sometimes 4 in., yet with only a single 1¼ in. return coming back. In such cases, a 4 in. return header connected to the two inlets is an unnecessary expense. Steel boilers usually have only one return inlet and in most cases with cast iron boilers the use of one inlet will be sufficient. Some cast iron boilers have side inlets on both sides and rear inlets with connections to both side inlets preferable for most effective circulation, and of course all split boilers require connection to both inlets in order to produce circulation from each half section to the opposite half section. Incidentally, the general admonition to avoid any use of a bull head tee can be disregarded in the case of wet return piping even as it sometimes can at other points in heating work.

Some boilers come with a lot of steam outlets, as many as six, presenting quite a problem as to how many of these are really needed in view of the expense entailed for multiple tappings. The theory that all of these should be used or else the manufacturer would not have provided such outlets is erroneous, because two successive sizes of the same type and make of boiler will often have the same number of outlets for boilers varying in capacity. This practice comes about due to convenience in manufacture and assembly, while the practice of providing a lavish number of outlets is merely one of those engineering contingencies against the chance of someone developing full nominal rating of the boiler; in other words, overloading the boiler to the extent where the maximum number of outlets are required in order to pass more steam than the boiler was designed to do at reasonable loads. A boiler with multiple outlets may also have the advantage of less expense on a replacement job, for a boiler set as conditions require, yet with one or more of these outlets coming directly below existing piping, avoids any alteration in such pipes.

Generally speaking, as many full size outlets should be used on a boiler as will pass the steam required at peak load at a velocity of no more than 20 feet per second. If the necessary mathematics is carried out, one will find that a convenient short rule is 1.2 sq. in. of boiler outlet area for each 100 sq. ft. of equivalent direct radiation at boiler outlets. Just which of the outlets should be used depends somewhat on conditions, but in general every other outlet would be used on a six outlet boiler.

The cold water supply to a boiler may be connected direct to the boiler, though this offers some possibility of boiler crackage at this point, especially with a careless attendant who fills the boiler too rapidly. Because of this possibility it is best to connect water supply to the return piping. In one case a job was discovered with the hot water supply connected to the boiler which of course taxed the hot water supply and wasted service hot water whenever the boiler was flushed out.

The writer cannot end this discussion on boiler connections without taking a parting shot at one remaining form of piping connection, that is, the connection of the safety valve to the boiler. In recent years this connection seems to have degenerated from standard practice in that it is now quite common, particularly on gas boilers, to find the safety valve connected from some part of the header piping, often with quite a bit of pipe and fittings between it and the boiler outlet. For years the writer was always taught that the safety valve should be threaded directly into the boiler outlet, but today he can find no underwriter of insurance who seems to be much concerned about its more remote distance from the boiler.

Some risers have long runouts in the basement or long offsets concealed in ceilings where their heat cannot be used to advantage, or where sufficient clearance for size and grade of pipe is difficult to obtain. In such cases the runout and offset need not be more than 1¼ in. as sized for runouts of radiators up to 55 sq. ft. in capacity, and sometimes a short riser can be conveniently offset with 1¼ in. connection to an adjacent long heat riser which serves a bath room on each floor above the basement, instead of running each riser all the way down to the mains.

Small kitchens are often heated by a 2 in. heat riser which is terminated just below the top floor, on which floor a radiator is installed due to the additional heat losses of roof exposure. In such cases, the riser should be vented as the air valve on the radiator is not alone sufficient for quick venting. Sometimes such a heat riser may be used to advantage as a distributing riser to radiators in the adjacent rooms on each floor, in which event the riser may be vented through the several radiator air valves.

In computing the amount of installed radiation, we count in the surface of heat risers on the one hand, but on the other hand ignore entirely the surface of exposed distributing risers, no doubt on the reasoning that distributing risers are that much of a safety factor, though their additional heating surface may become an unsafe factor if located in the same room as the thermostat. In computing boiler load, the heat effect of distributing risers is taken into account in the usual percentage allowance for piping tax, but heat risers need no allowance for piping tax. The square feet of surface in the heat risers should be added to the load after the piping tax has been applied to the radiators.

In checking the heat requirements of an existing system, one should take into account the additional heat effect of all distributing piping before condemning the radiators as insufficient in size. The writer is reminded at this point of a well known skyscraper building in which the available space was not sufficient for the length of radiators as computed necessary for the special type of radiation in mind. The radiator sizes were therefore reduced to fit upon guarantee of the manufacturer as to their heat adequacy and they are still satisfactorily heating the structure after a lapse of many years, merely because of the fact that each radiator was backed up by an additional 10 sq. ft. of surface in the exposed steam and return runouts for each bay.

Survey Shows 42,604 Air Conditioning Jobs

THE 1940 survey of air conditioning installations in the United States, broken down into business classifications and just released by C. E. Greenwood, commercial director of Edison Electric Institute, is the most comprehensive study of air conditioning progress ever made. The 183 utility companies represented in the report serve 75,000,000 people, or more than half of the country's total population. The number of installations reported is 42,604, which use a total of 873,024 hp.

One-fourth of all these air conditioning installations are in offices and office buildings; one-fifth are in restaurants and retail stores, and one-fifth are in apartments. The remainder are scattered among nearly two dozen different commercial and industrial classifications, including banks and barber shops, beauty parlors and broadcasting stations, hospitals and hotels, funeral homes, public and recreational buildings, department and drug stores, theatres, manufacturing plants of many kinds, and fur storage establishments.

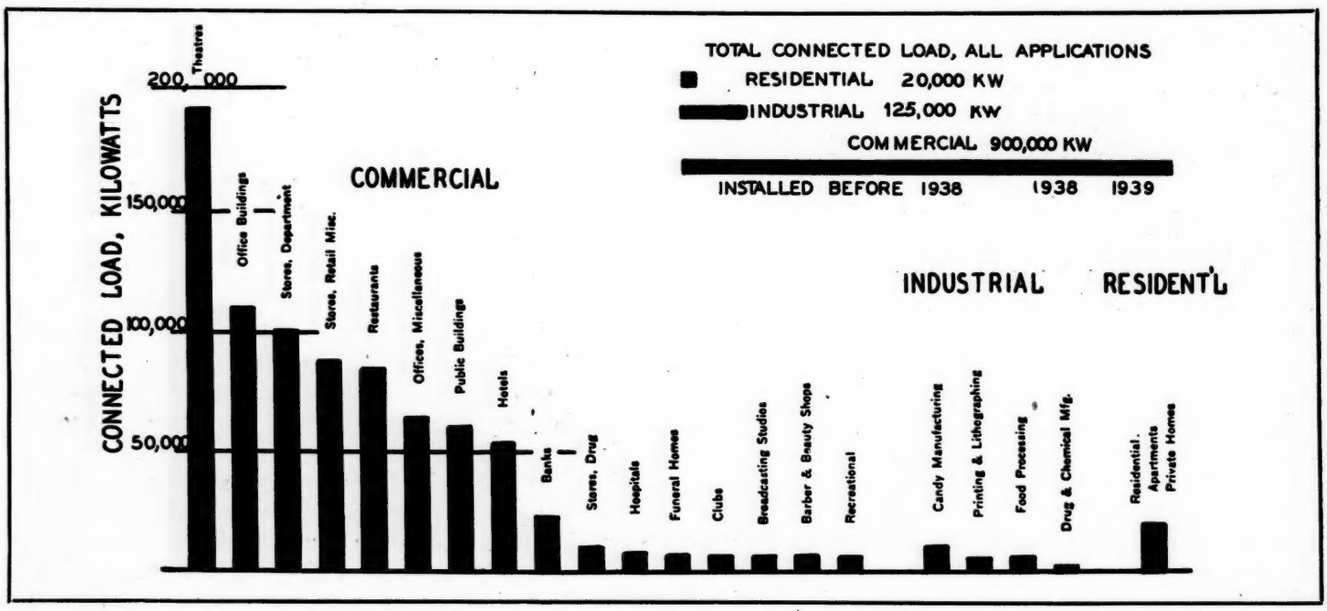
Data for each utility company is shown for 30 different business classifications, and includes a record of installations prior to 1938, during 1938, during 1939, and total installations to the first of January, 1940.

Variations in the average horsepower of air conditioning systems for various types of service is conspicuous throughout the report. The average doctor's or dentist's office, for example, uses only 2 hp. The average public building installation requires 142 hp. to operate its air conditioning system.

There are ten times as many doctors' and dentists' offices with air conditioning as there are air conditioned department stores, yet the latter, using an average of 109 hp., account for a total of 58% more horsepower than this group of offices. Other heavy users include theatres, hotels, office buildings and retail stores.

AIR CONDITIONING INSTALLATIONS IN THE U. S.
(As reported on the lines of 183 electric utilities as of Jan. 1, 1940. From the 1940 survey of Edison Electric Institute)

Application	No.	Hp.
Residential: Apartments, Private Homes	9,664	17,348.90
Commercial:		
Apartments	82	2,466.65
Banks	526	19,777.80
Barber and Beauty Shops	645	4,205.98
Broadcasting Studios	156	4,825.80
Clubs	268	5,039.07
Funeral Homes	683	6,353.73
Hospitals	420	6,684.18
Hotels	1,087	45,644.31
Office Buildings	974	96,054.02
Offices, Doctors and Dentists	1,360	3,193.26
Offices, Miscellaneous	8,308	55,536.72
Public Buildings	363	51,603.10
Recreational	142	3,007.76
Restaurants	4,518	73,255.17
Stores, Department	806	87,899.64
Stores, Drug	785	9,406.96
Stores, Retail Miscellaneous	4,889	76,658.57
Theatres	1,942	167,280.92
Other Commercial	1,861	29,373.19
Industrial:		
Candy Manufacturing	181	10,009.92
Drug and Chemical Mfg.	60	2,949.80
Printing and Lithographing	112	6,848.96
Textile Manufacturing	43	2,287.00
Tobacco Manufacturing	27	1,500.43
Metal Working	9	325.00
Food Processing	268	6,035.54
Fur Storage	120	637.61
Other Industrial	2,190	74,472.94
Miscellaneous Industrial Not Classified	115	2,341.15
Total Installations	42,604	873,024.08
Installations included above that use power other than central station electric	263	8,799.21
Self-contained conditioners included above:		
1½ hp. or less (Room Coolers)	10,786	7,645.13
2 hp. or larger (Store Units)	2,689	13,297.59



Connected air conditioning load on lines of 183 utilities as of Jan. 1, 1940.

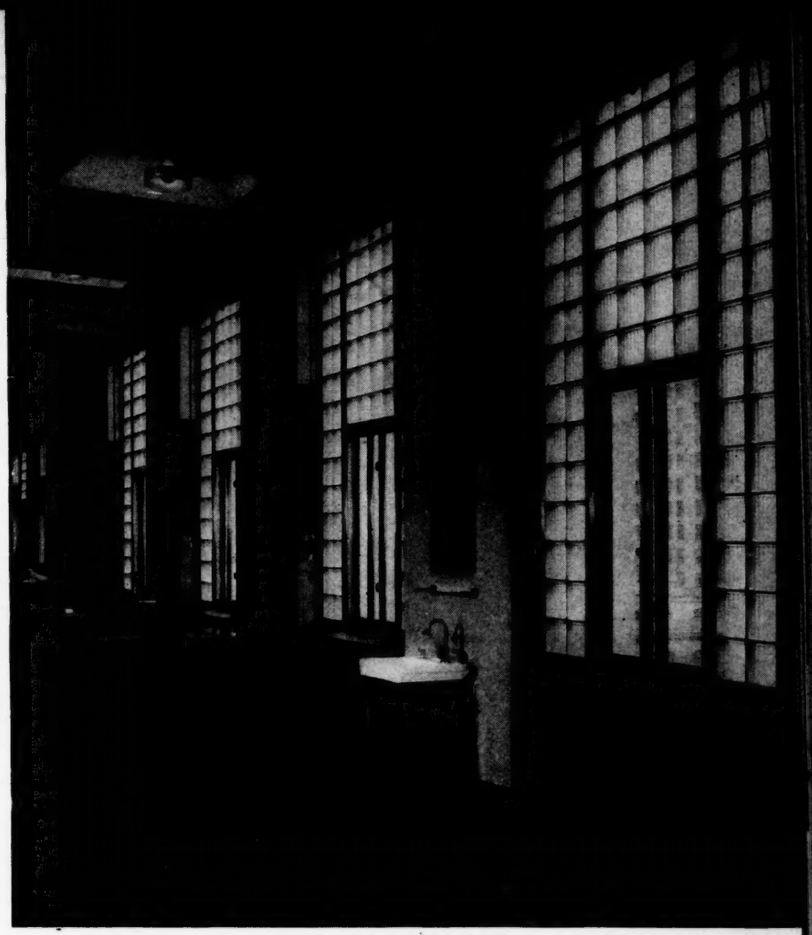


Fig. 1. Two of the sectional steel wall panels have been removed to show the copper piping of the wall warming system. By circulating hot water through this system, local sensations of warmth or coolness from interior partition to outside wall are practically eliminated. Exterior wall and window temperatures are kept at a uniform level and thermal losses through the glass or the sectional steel paneled, cork-insulated walls are balanced.

Wall Warming System Neutralizes Glass Loss in Office Building

WHEN new buildings are erected today, the experience and the advances of the last few years in modern air conditioning and heating are being definitely recognized. Nowhere is this more pointedly demonstrated than in the unusual new home office building of the Bankers Life Company, one of America's ranking legal reserve life insurance companies, located in Des Moines.

This handsome structure consists of six main floors, two basements, and a seventh floor penthouse containing the directors' room and part of the air conditioning equipment. So many innovations have been incorporated in its construction that it has been called "the building of the decade."

One of the major factors contributing to this recognition is the combination of air conditioning and a unique wall warming system, both of which were designed by Charles S. Leopold, consulting engineer of Philadelphia. Mr. Leopold was called in by the architects—Tinsley, McBroom & Higgins of Des Moines.

A system with a capacity of 650 tons accomplishes the task of year-round air conditioning, with two compressor units of 260-ton capacity and a smaller unit supplying the additional 130 tons. Powered by 4,000-volt synchronous motors of 300 hp. for the large units and 130 hp. for the smaller compressor, this is one of the largest systems yet installed west of the Mississippi.

It is interesting to note that the air conditioning system is always a cooling system, in winter as well as summer. The air supply comes through the ceilings always at room temperature or below, with adequate provision for the additional heat introduced by the occupants and the lighting.

To be sure, this system could be used to heat the building. But the consulting engineer recognized the problem characteristic of the Iowa climate, with its extremes of heat and cold. During the heating season there is usually a sensation of cold near wall and window areas of the ordinary building, caused by descending currents of cool air at the exterior walls and by radiation of body heat to the cold surfaces of walls and windows.

Consideration of this problem led to the development of the wall warming system, a new departure that is bound to be of interest to heating and ventilating engineers and contractors.

With all masonry walls insulated by a two-inch layer of cork, a system of one-inch bare copper piping loops around and under the windows, carrying hot water under forced circulation. (In summer, chilled water circulates through the piping, but it was not considered advisable to increase the surface to compensate for full heat transmission and sun effect under summer conditions.)

The lining of the interior walls consists of sectional steel wall panels, the first installation of its kind in a large office building. These removable panels, factory-enameled in pleasing shades of grey-green and ivory, will have an upkeep cost only 20% as great as that of the traditional materials.

The wall warming system, then, operates between these steel panels and the cork insulation. Its purpose is to balance any thermal losses and achieve uniform wall and window temperatures that will limit the radiation of body heat.

Radiant heating as it is generally known provides for

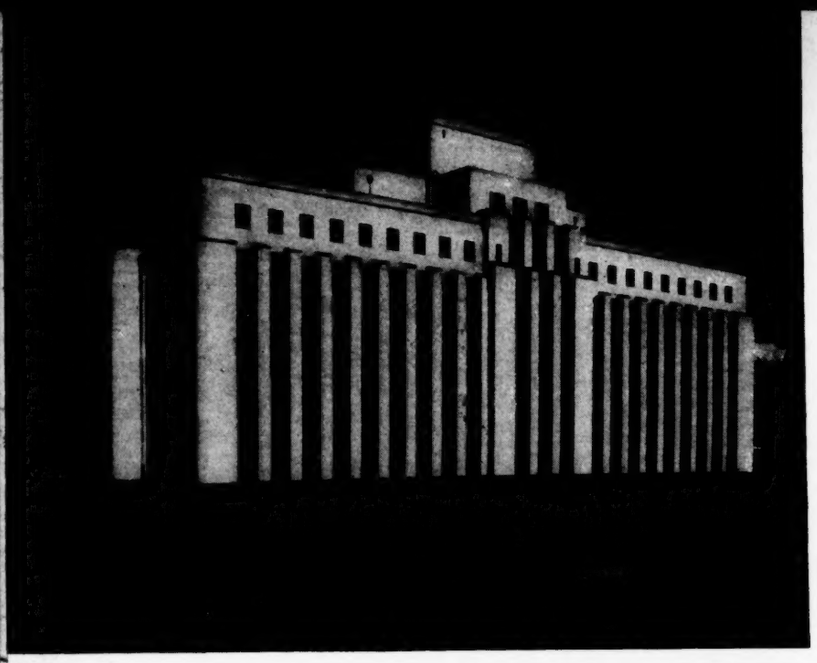


Fig. 2. Home office building of the Bankers Life Company in Des Moines, Iowa, which has one of the most modern and largest conditioning systems west of the Mississippi.

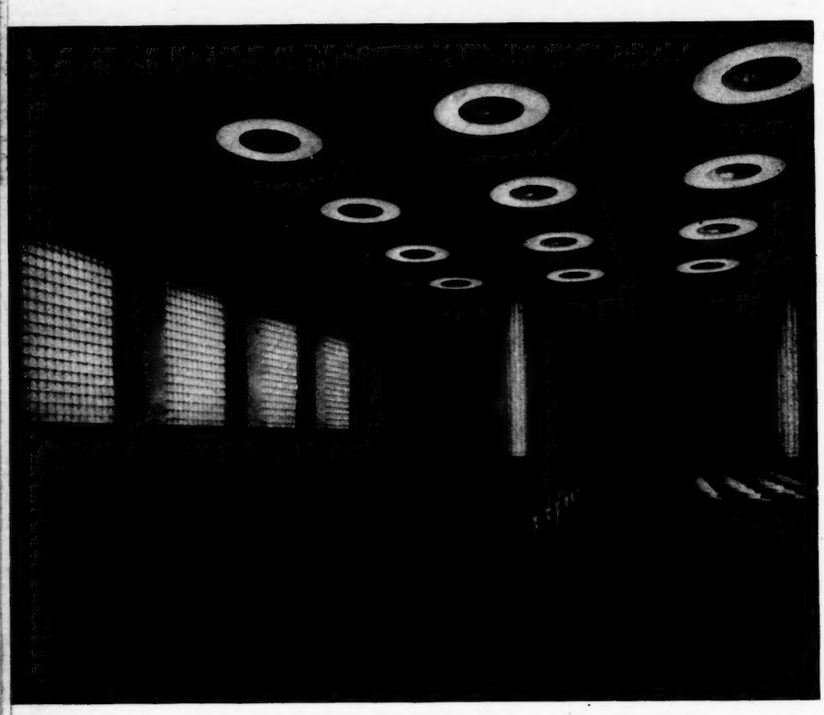


Fig. 3. This view of the Bankers Life Building auditorium which seats 1100, shows the combination of lighting units and high velocity air diffusers in nine-foot ceiling.

the heating of the building surface to temperatures of more than 100F. In the Bankers Life building, it is only necessary to keep wall and window temperatures in the neighborhood of 70F. The efficiency of the wall warming system in this respect is indicated by thermocouple tests. The widest temperature variation of the wall from floor to ceiling was 3F, with the highest temperature occurring directly beneath the window. This check was made at a time when the air conditioning system was not in operation.

With heat loss through the walls and windows balanced by this system, it is easy to see how the conditioned air introduced through the ceilings can be kept at a predetermined level and serves as a cooling medium throughout the year.

Panels of glass block surround the double-glazed windows, with the storage of radiant heat in the glass brick reduced to a minimum by shading from projecting piers.

Three low-pressure oil burning boilers of 185 hp. each supply the steam required for heat exchangers, convectors, and domestic hot water. Bunker C oil for the burners is stored in three tanks with a combined capacity of 16,500 gallons.

Control panels in the refrigeration room govern the operation of all ventilating equipment. Selector switches, in connection with an electrical thermometer, tell the operator immediately about temperature conditions at key points throughout the building and in the apparatus. A six-point recorder keeps a running record of temperatures whose trend is helpful in the operation of the plant.

Similar control equipment in the boiler room indicates pressure, over-fire draft, and degree of smoke, with combustion recorders mounted on a separate panel.

In the building superintendent's office, the indicating portions of these controls are repeated. Also, there is an instrument on his board to indicate the smoke of the total chimney, and an alarm to notify him of an objectionable smoke condition.

Visitors to the building can stand on an observation platform and look downward through a large plate-glass window at the equipment in the compressor room. This continues the Bankers Life Company program of encouraging the public to share the interest in the "building of the decade."

Fig. 4. In the building superintendent's office the control board contains a combustion recorder, dials indicating the fuel oil supply in the storage tanks, and flow meters for chilled water and condenser water. The panel at the lower right contains a master thermostat and pilot lights which show the operation of fans, spray pumps, compressors, radiant heaters, booster heaters, boilers, and Precipitron. The corresponding panel at the left contains selector switches which, in conjunction with the electrical thermometer immediately below the clock, enable the superintendent to check temperatures at key points in the apparatus and at 24 stations throughout the building. Below the electrical thermometer is a recorder which indicates the trend of the following temperatures: outside wet bulb, outside dry bulb, apparatus dewpoint, fan discharge, and return air.

FHA Puts Pressure on Modernizing; Big Campaign Breaks This Month

NO branch of the building industry benefits more from modernization loans insured by Federal Housing Administration than does the heating industry. According to the Division of Research and Statistics of FHA, cumulative figures to the beginning of the present year show that 32.2% of the number and 27.7% of the amount of loans for property improvements insured under Title I of the National Housing Act are for improvements to heating plants. This far outstrips any other classification of building; the next highest percentage in amount is 17.4% for exterior painting.

In view of this and also due to the fact that the heating industry's boom months are during the fall, the heating industry will be keenly interested in the big drive planned by FHA on property modernization which will break simultaneously in all sections of the country in mid August. The new drive, patterned closely after last summer's similar and highly successful campaign, will again stress "Fix Up Your Home! Modernize", and convenient monthly payments on the FHA Plan of installment buying will be featured.

Manufacturers and dealers who tied in aggressively to last year's drive found that many modernization jobs can be closed and new prospects developed by featuring FHA-insured (Title I) loans.

Has all the property improvement already been done? FHA officials think it's only begun, since applications for loans seem to be steadily rising and at present average about 10,000 a week. This increased activity indicates that the public and the lending institutions are coming to realize and avail themselves of the benefits offered by the FHA Plan. The mounting figures also suggest that the building industry would do well to engage in some concerted stimulating action of its own during the coming months.

As a matter of fact, FHA officials throughout the country point out that industry is approaching that agency more and more frequently with cooperative suggestions and plans to make the most of the FHA modernization and repair program. In line with these suggestions, FHA is again preparing literature and display material for the trade. This will be available at headquarters in Washington, or through FHA's 64 field offices, about the middle of August.

The material includes a colorful window display printed in dark red, blue, and yellow and a cleverly illustrated booklet with a check list for home owners, printed in red and black. A mimeographed piece will describe how dealers can develop and handle business under the "Property Improvement Credit Plan", Title I of the National Housing Act. It is a brief merchandising manual for dealers suggesting means by which they may increase their sales through the FHA plan of installment payments.

Both the display material and literature are available to all dealers wishing to be identified with the national modernization program, but it will be sent only on direct request. To avoid duplication, manufacturers and trade associations should have the dealers themselves send in their orders. It is suggested that this be done as soon as possible since quantities are limited and it is a case of "first come first served," in filling requests.

The window display pieces are designed to be used separately and may be adapted to the needs and display facilities of individual dealers. However, this material is essentially a unified window display in which all pieces are employed to feature the dealer's merchandise. A colorful streamer bearing the legend "Modernize—Repair" extends across the window, and side streamers suggest improvements to "Fix Up Your Home." In order to suggest to dealers how to arrange this material attractively in their windows, a descriptive folder has been prepared. Copies are available for manufacturers, trade associations, and others who wish to tie in their sales outlets with the national modernization program.

Specimen newspaper advertisements are also prepared by the Federal Housing Administration for dealer use. These are mailed in proof form to local newspapers who will be supplied with mats of the art work on request.

Other media besides store display, direct mail, and newspapers will be used during the forthcoming campaign. A car card will be released for national distribution in trains, cars, and buses.

Special radio programs are being arranged locally by State and District Offices of FHA. In addition, manufacturers, dealers, and others will be provided, upon request, with scripts and commercial announcements by which their products may be tied in to the national program for Home and Business Property Modernization. A special transcription of dramatized radio "spots," suitable for dealer tie-in, will be made available through local radio stations.

Determining Minimum Requirements of Insulation

By ROBERT K. THULMAN‡

The author proposes minimum requirements for insulation, based on the idea that insulation should be considered as an integral part of the wall construction and that the primary objective should be comfort as it is affected by wall surface temperatures. This article is abstracted from a paper presented at the Centennial Meeting of the National Mineral Wool Association held at Chicago, May 16, 1940

SUCCESS might reasonably be expected from a minimum standard for insulation if there can be found a basis for the insulation standard which is arrived at from the standpoint of the function of insulation as an integral part of a wall.

The first consideration is based on the question of comfort as it is affected by wall surface temperature. Just what this minimum wall surface temperature should be is not now apparent. It can be developed by research. It is not a new discovery that comfort is measured by other factors than just a dry bulb temperature which, for this country, is generally agreed to be between 70F and 75F, but it has only lately been appreciated that the mean radiant temperature of the surroundings has a good deal to do with comfort.

The present tables of beam sizes found in building codes, specifications and minimum construction requirements were developed as a result of research which concluded that a deflection greater than 1/360 of the span was excessive. Similarly it should be comparatively simple to discover at what mean radiant temperature comfort within a house falls below a certain desirable standard.

The research work necessary to determine the minimum wall surface temperature should be done under conditions that closely approximate actual conditions determining the mean radiant temperature of the average house. It would not appear wise to ignore the tendency to use larger glass areas than has been the habit in the past and it will probably be necessary to discover the effect of varying single and double glass, external wall, ceiling and floor areas. Solely as a means of demonstrating the principle, a wall surface temperature of 65F and an inside air temperature of 70F have been selected in preparing the charts. Purely a random figure, convenient to use, 65F may be found to be either too high or too low. Scientific investigation will have to be undertaken to determine it. The charts are based on a surface coefficient of 1.65 B.t.u. per sq. ft. per hour per degree temperature difference and are calculated only. No actual measurements or tests were made in the laboratory.

The charts consider two conventional types of construction—the frame wall with a coefficient of .26 and the masonry wall with a coefficient of 32. The values shown are based on maintaining an inside temperature of 70F. Chart 1 shows the wall temperature occurring with various outside temperatures for these two walls.

Chart No. 2 shows the coefficient "U" required to maintain a wall temperature of 65F for outside tem-

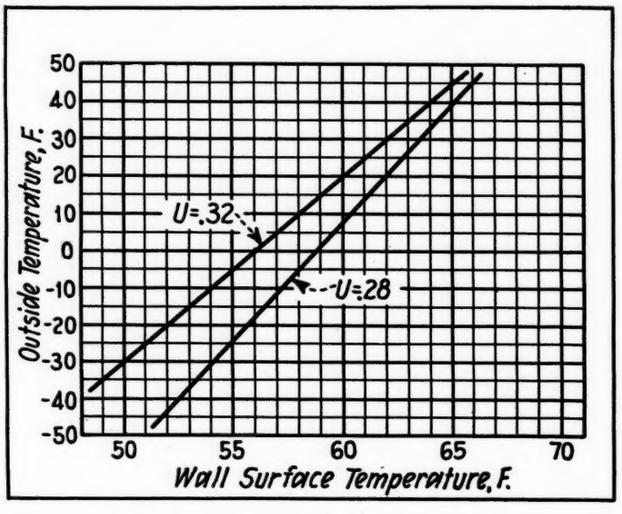


Chart 1.—Wall temperature at various outside temperatures for two types of construction.

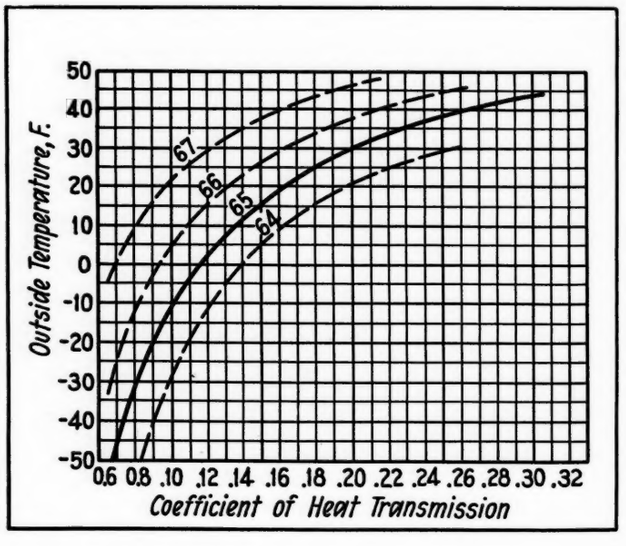


Chart 2.—Coefficient U required to maintain a given wall temperature at various outdoor temperatures.

‡Mechanical Engineer, Federal Housing Administration, Washington, D. C.

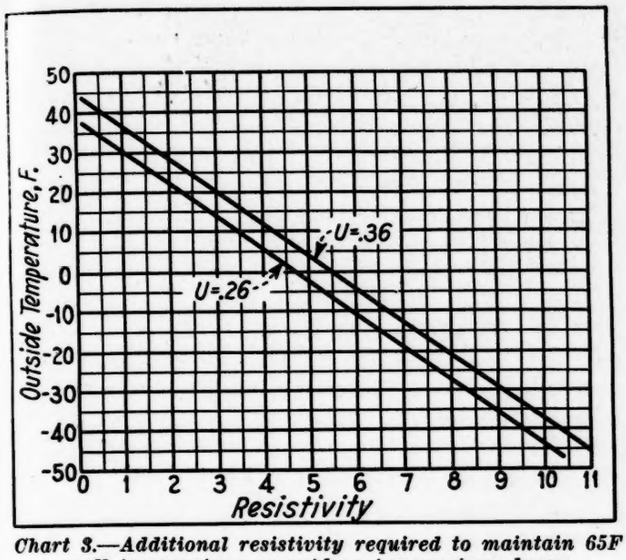


Chart 3.—Additional resistivity required to maintain 65F wall temperature as outdoor temperature drops.

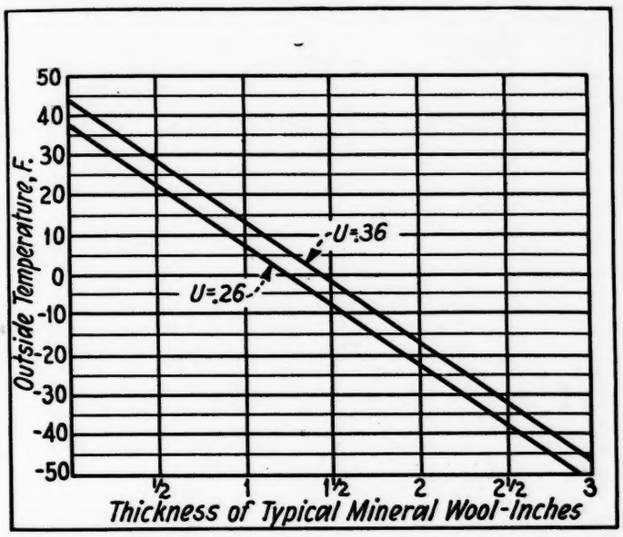


Chart 4.—Additional insulation required to maintain 65F wall temperature as outdoor temperature drops.

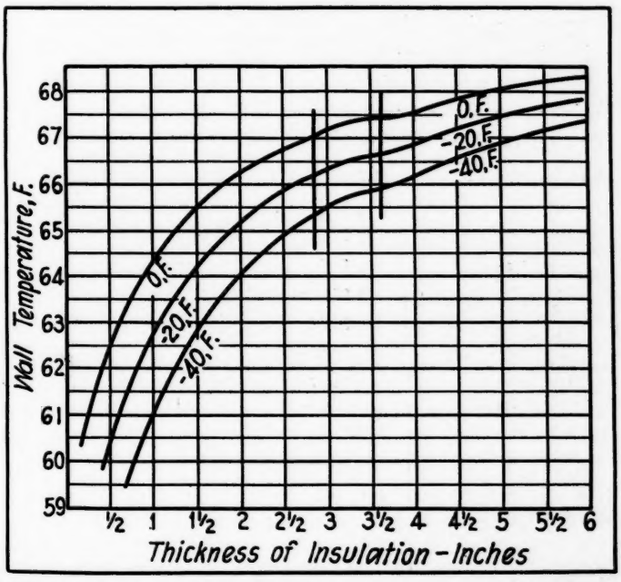
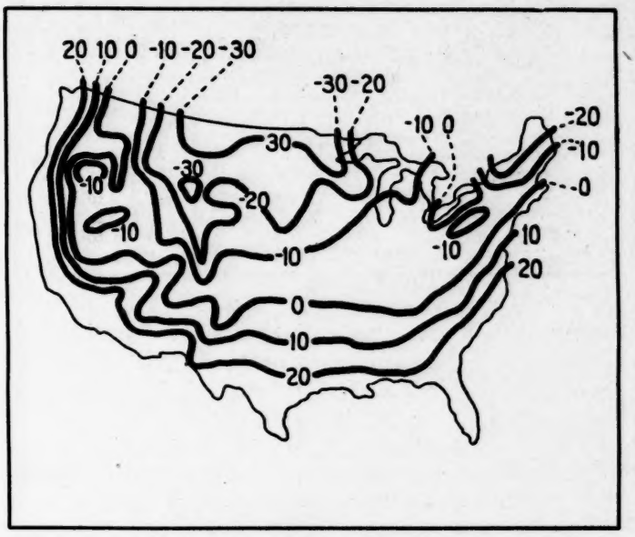


Chart 5.—How added thickness of mineral wool increases wall temperature for three selected outside temperatures.



Map 1.—Outside design temperatures for the United States.

peratures down to —50F. The three broken-line curves are for surface temperatures of 64, 66 and 67F, and show what a marked influence a change of only a single degree has on the required coefficient.

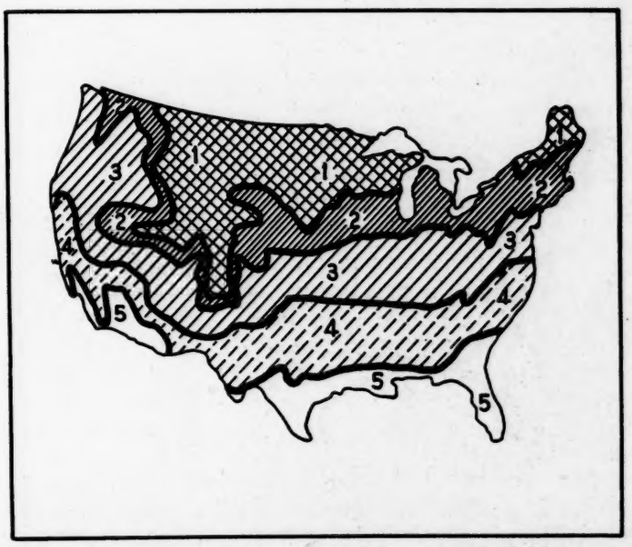
Chart No. 3 shows the additional resistivity required for each of the two conventional walls to maintain the wall surface temperature at 65F under varying conditions of outside temperature.

Chart No. 4 is the same as No. 3 except that the resistivity is changed to insulation thickness.

It should be emphasized again that a 65F temperature for the wall surface was selected arbitrarily to demonstrate the approach and is not to be construed as having been arrived at on a scientific basis.

Chart No. 5 shows how added thickness of mineral wool increases the wall surface temperature for three selected outside temperatures. Between 2⅞ in. and 3⅝ in. the curves flatten out because of the gradually reduced value of the narrowing air space.

The amount of insulation necessary to maintain a given minimum wall surface temperature depends, ob-



Map 2.—Zoning of the U. S. based on the degree-day load and design temperature.

TABLE 1.

Designation of Zone Numbers According to Degree Days and Temperature Difference

Degree Days	Minimum Temperature, F.	Temperature Difference, F.	Zone
8,000 to 10,000	—20 to —30	90 to 100	1
6,000 to 8,000	—10 to —20	80 to 90	2
4,000 to 6,000	0 to —10	70 to 80	3
2,000 to 4,000	+10 to 0	60 to 70	4
Below 2,000	+20	40	5

viously upon the outside temperature. The basis for determining the outside temperature must necessarily be considered in the development of the standard on which the requirement is based.

The outside temperature might be the same as that commonly used in determining the required capacity of a heating plant. This temperature is, in general, 15 to 20F higher than the lowest ever observed. Design temperatures for heating systems are shown graphically in Map 1.

A probable compromise, however, might consider both the minimum design temperature and the heat requirements in terms of degree days.

A zoning of the country in this manner (Map 2) has been made in accordance with Table 1 and Map 1.

The results of a minimum requirement for wall insulation based on wall surface temperature are manifold and may be far reaching. They may be divided into two kinds.

The first of these two kinds of results has to do with money. The requirement disposes of such questions as the length of the amortization period, return on the investment, and other uses for capital. Most important, however, is that, properly applied, the requirement should result in a substantial reduction in the cost of heating, a lower cost of home ownership and a better mortgage risk.

The second group of results is physiological. Higher wall surface temperatures should tend to maintain the normal ratio of heat losses from the body in a proper equilibrium. In so doing they should assure greater comfort.

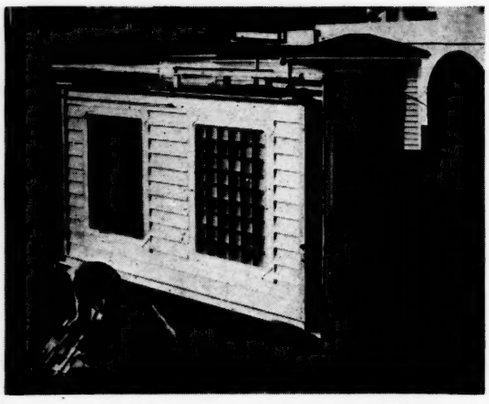
Of immediate importance to the insulation industry is that a requirement of this nature first disposes of the question of how much insulation should be used, and, second, and more important still, may serve to refute the popular conception that insulation has to pay its way in cash savings in order to be worth while. This standard may serve not only to refute this popular notion but if scientifically established may, instead, instill the idea in the public's mind that insulation must be included in a wall solely to maintain an accepted minimum standard of comfort.

Heat Flow Through Glass Blocks Studied

Old Sol has given the air conditioning engineer an intricate problem where walls of glass block construction are specified. The question is—how much additional cooling capacity must be provided in an air conditioning system to compensate for the solar heat transmitted through glass blocks? To assist the engineer in obtaining such fundamental engineering data the Committee on Research of the ASHVE outlined a research program utilizing a miniature house as shown in the accompanying photo for investigation at its own Laboratory in Pittsburgh.

The structure is essentially a large heavily insulated test cubicle mounted on a turn-table so that the glass block panels may be oriented to the sun in any direction. The cubicle is divided into two equal compartments by a solid partition between the identical glass block panels. There is a door in the rear of each compartment for access to instruments and the introduction of ice.

Transmission of heat through each glass panel was measured by placing ice in each compartment and then measuring the quantity of ice melted. One glass panel was shaded while the other was allowed to receive the full effect of old Sol's rays. The difference between the ice meltage in the unshaded compartment and the shaded compartment gave the measure of the sun heat transmitted by the unshaded glass block panel. In addition to the amount of ice meltage, measurements were made of intensity of sunshine before and after it had passed through the unshaded glass block panel, the temperature of the air inside and outside the compartment, the temperature of the face of the panels, wind direction, wind velocity and other attendant conditions.



In addition to maintaining its own Research Laboratory where the glass block tests were conducted, the Society's Committee on Research of which A. E. Stacey, Jr., is chairman, is actively sponsoring cooperative investigations in several colleges and universities and at the present time research work of this character is in progress at twelve institutions. These research studies deal with fundamental engineering problems such as frictional flow of hot water in large size pipes, corrosion in steam heating systems, friction of air in ducts, air distribution from inlets and outlets, performance of radiators and convectors, sound transmission in ducts, efficiency of air cleaning devices, fundamental properties of cooling towers, deviations of air-water mixtures from laws of perfect gases and physiological reactions of people to different environments.

CANDY

INDUSTRY

EDGAR P. LEWIS & SONS, INC.

large-scale manufacturer of candy, Malden, Mass., has improved the appearance of its product and made material savings in time and materials through the use of ten Bryant silica-gel type air dehumidifiers.

By DANA R. McNALLY†

ON an average working day two tons of water is being removed from the air in the production and storage rooms of the Malden, Mass., plant of Edgar P. Lewis & Sons, Inc., large scale manufacturers of fine candies. This moisture is removed by ten gas fired silica-gel dehumidifiers operating independently of any of the other existing functions of air conditioning, being controlled directly by humidistats located in the conditioned spaces.

The gratifying results of this extensive use of drying equipment have been an improvement in the appearance of the candy and material savings in goods and time.

About two years ago a silica-gel dehumidifier was installed in two storage and packing rooms of the Lewis plant where a condition of 65 to 70F at 40% to 45% relative humidity was maintained. This machine was not applied to previously unconditioned spaces, but to rooms which had been considered adequately air conditioned with thirty tons of refrigeration furnished by two compressors. It was through experience with this one installation and from the knowledge there gained of the effect of dehumidification on production and packing of candy that this particular company has made the installation of the additional nine dehumidifiers.

An interesting point reported by engineers of the Lewis organization is that low temperatures previously believed to be required in the candy business are not always necessary. Many production troubles which had been attributed to too high temperatures ironed themselves out when the proper relative humidity was maintained. Then, too, it has been found that a higher temperature with properly controlled humidity is sometimes a condition more desirable to certain processes of candy making than those low temperatures felt absolutely necessary a few years ago.

Hard Candies and Jellies

One of the silica-gel machines is being used on a room 100 ft. long, 40 ft. wide, and 12 ft. high in which

†Industrial Appliance Co. of New England, Boston, Mass.

THIS INSTALLATION

Owner—Edgar P. Lewis & Sons, Inc., Malden, Mass.

Contractor—
W. Warren Cummings and Associates, Boston.

PROFITS IN AIR

NO. 24 OF A SERIES DEALING WITH ATMOSPHERIC CONDITIONS IN INDUSTRIAL PLANTS.

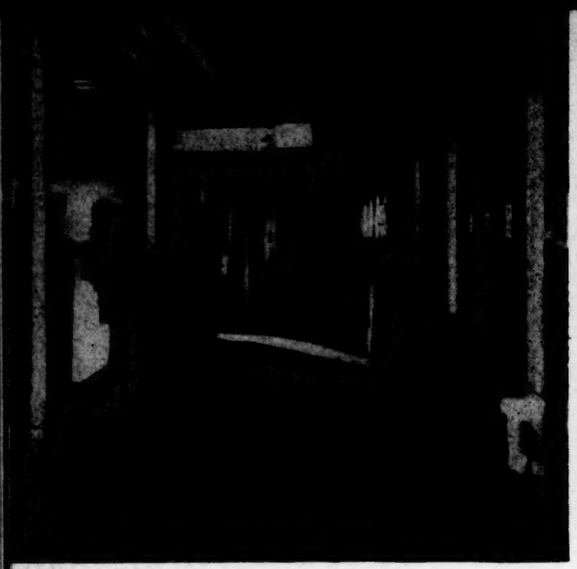


Fig. 1. Operations in this hard candy and jelly packing room were formerly often entirely suspended because of high humidity. Since installing the dehumidifier, this department can be depended upon to operate regardless of weather.

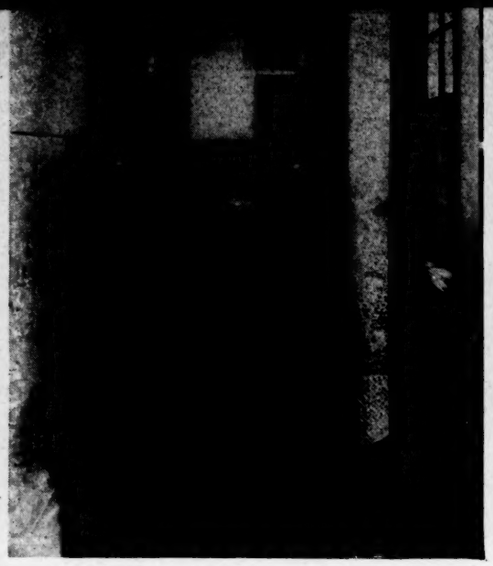


Fig. 2. This tray type silica gel machine automatically feeds dry air to the refrigeration unit in the hard candy and jelly packing room thus enabling the space to be kept at a constant condition of 75F dry bulb and a 45% R.H. at any outside condition.

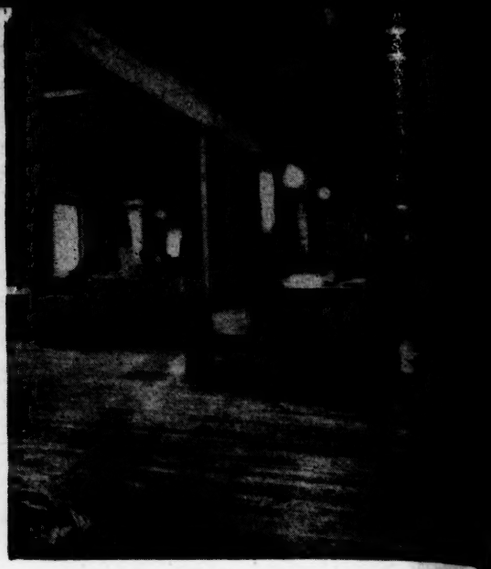


Fig. 3. Cellophaning machines operating in this packaging room are vulnerable to high humidities as well as low. The installation of two dehumidifiers in conjunction with well water has eliminated expensive tie-ups in production at this point.

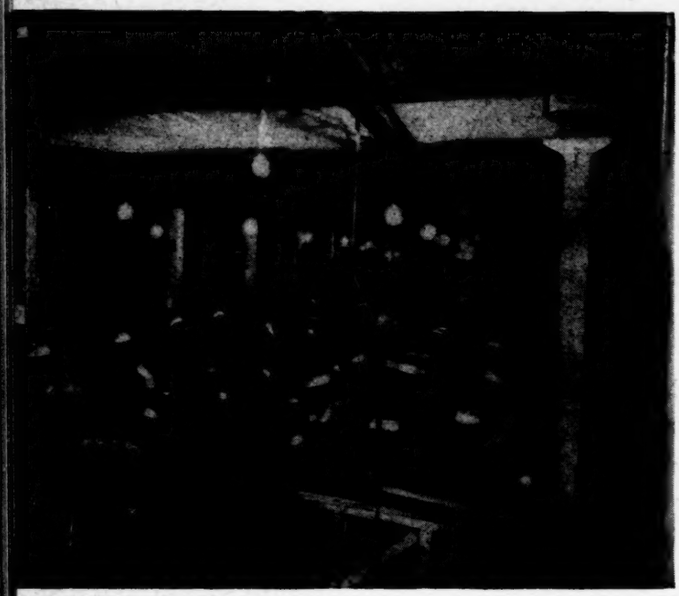


Fig. 4. In this hand dipping room are sometimes as many as 400 girls working at top speed in a space 100 x 132 x 12 ft. The value of independently controlled relative humidity in this space is recognized to the extent that it is guarded by the largest size silica gel dehumidifier.

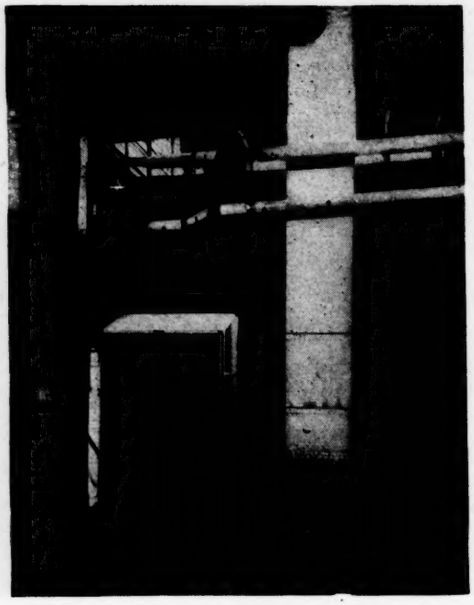


Fig. 5. The tray type dehumidifier which, operating in conjunction with refrigeration, has eliminated the necessity of sending the hand-dippers home on days when the humidity is high outside. Two dehumidifiers maintain 63F and 50%.

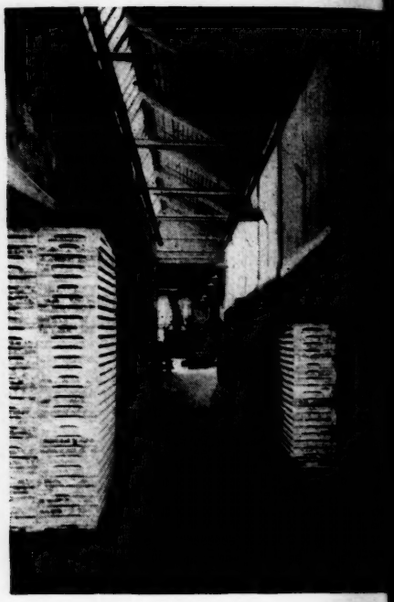
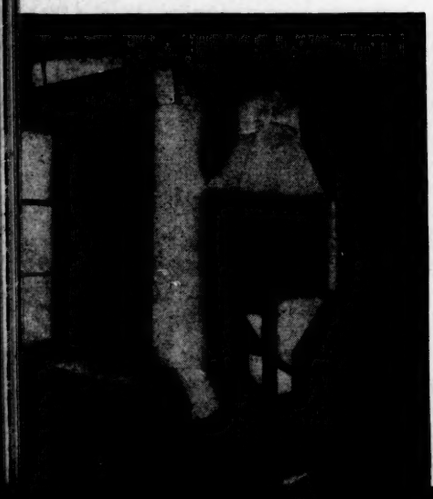


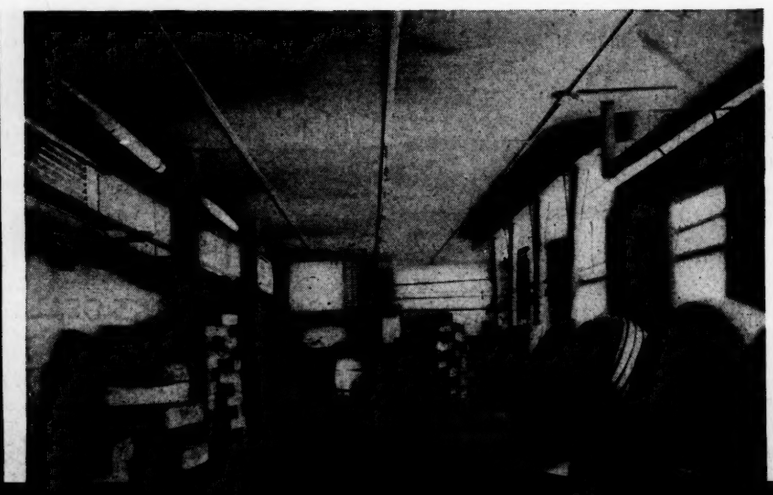
Fig. 6. These starch rooms were satisfactory except for caramel or molasses base pieces in humid weather. To prevent tying up the packing of entire assortments, a dehumidifier is now employed.

Fig. 7. This rotary type silica gel machine supplies dry air to the starch compartments where it facilitates drying.

Fig. 8. Refrigeration and dehumidifying together have been installed in this nut and pan room. The drying of these goods is satisfactory and the drying time is a predetermined interval. A 60F temperature and a 40 per cent relative humidity are independently maintained.

Fig. 9. This rotary type silica gel machine has proved that 60F dry bulb with a 40% relative humidity is the answer to the nut and pan room drying problems.





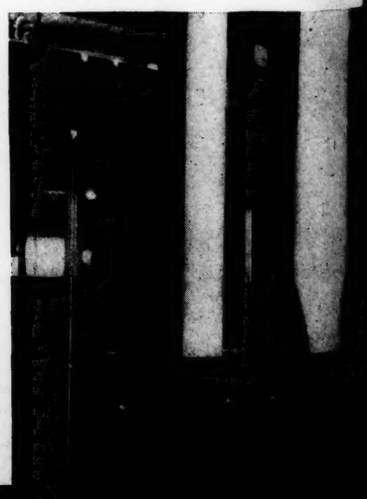


Fig. 10. These tray type silica gel machines supplying automatically dried air to an extended surface cooler using 54F well water, maintain the cellophaning room at 70F, with 50% R.H.

Fig. 11. An enrober room in the Lewis plant, having a cubage of 130,000 cu. ft. and occupied by 300 employees, has been materially improved by the addition of a large size gas fired dehumidifier. Although two of these machines were originally specified for this space, the one now installed successfully handles the job except in most severe weather.

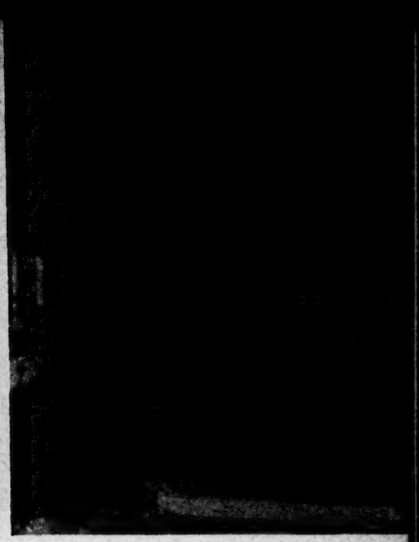


Fig. 12. The silica gel machine which operates with 60 tons of refrigeration to produce satisfactory conditions in the enrober room.

hard candies and jellies are packed. This room was not specially constructed for the application. It was previously cooled by a six-ton refrigeration unit. There are about 100 employees in the space and a condition of 75F and 45% relative humidity is maintained. This is one of the spaces where the engineers are no longer particularly fussy about the temperature maintained, but they are most particular about the humidity curve.

With the present method candy comes off from the conveyor and is caught in trays, carried into the packaging room, and emptied onto the mixing table, layer on layer, and then whenever convenient, mixed and packed. If some remains on the table over night or over the week-end, no one is concerned, because the candy cannot become tacky or sticky when the humidity is controlled.

Cellophane Room

Two silica-gel machines with well water at 54F are used to condition the Cellophane Room. This is a space 90 x 48 ft., with a 12 ft. ceiling. This area was not conditioned before, and some refinishing was done to make the application more practical. Interior partitions are constructed of two layers of insulation board and double windows were installed. A condition of 68 to 70F with a relative humidity of 50% is maintained. This installation is particularly satisfactory because previously there were many difficulties encountered with the cellophane machines which have now, by controlling the humidity, been eliminated.

An interesting feature of this job is that the water which is run through the coils to cool this room is also used afterwards to cool the compressors which are used on other spaces. This nets a surprisingly low operating cost for the conditioning of this room and made for a low initial installation cost.

Enrober Room

The Enrober Room, where candy is coated with chocolate, a space 100 x 100 x 12 ft., with an ell 30 x 32 x 12 ft., is also served by a silica-gel machine. It was calculated that this area required much more dehumidification than is actually installed; nevertheless, the installation has succeeded in greatly improving conditions. There are about 200 employees in the room which, with its rather large volume, constitutes a fair-sized load. There was no attempt made here to either insulate or in any other way cut down moisture or heat gain. Sixty tons of refrigeration was used previously to condition this area and still handles the cooling load.

Storage Room

Another storage space, 100 x 45 x 12 ft., was conditioned with two silica-gel dehumidifiers this past season. The fifteen-ton mechanical cooling unit had already been installed and is used to maintain the temperature requirement of 66F. A relative humidity of 50% is desired here and is being accurately held there by the two gel machines. This installation differs from the other installations in that each of these machines is controlled independently by its own humidistat located in each end of the room. Since there are only about half a dozen employees in the space the moisture gain is almost entirely due to infiltration.

Nut and Pan Room

The results obtained in the pan room after the installation of the silica-gel dehumidifier have been gratifying. Formerly a comparatively low temperature and high humidity were maintained. Now a 60F temperature and 40% relative humidity are independently held by the gel unit, so that the drying time is always satisfactory regardless of weather conditions.

Hand-Dipping and Packaging Room

The large hand-dipping and packing room where 300 employees work, 135 of them hand-dippers, is 100 x 132 x 12 ft. This room has double glass windows. It was estimated to require two silica-gel machines, but one was installed for the present and has been entirely

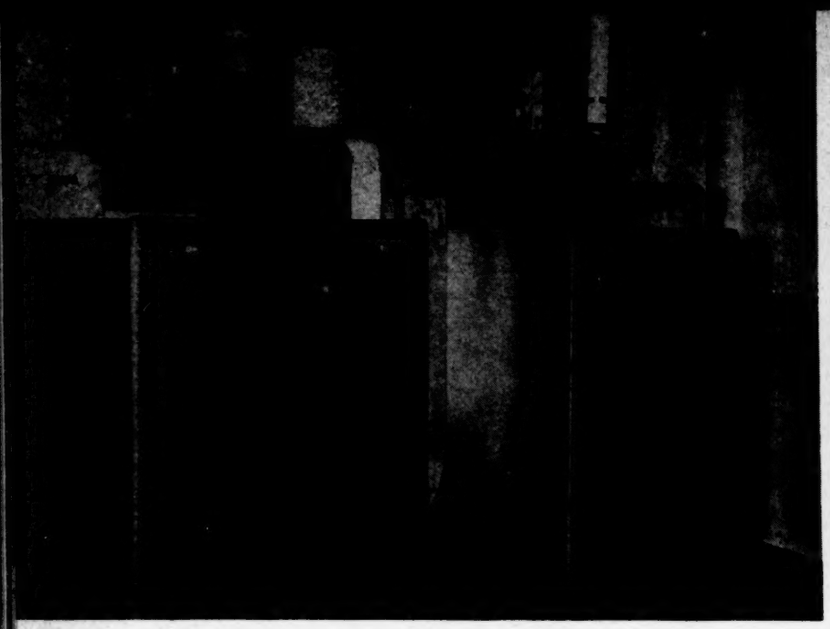


Fig. 13. These two tray type silica gel machines operate in conjunction with a 15 ton refrigerating unit to maintain the storage space at 66F with a 50% relative humidity, independently controlled and maintained.

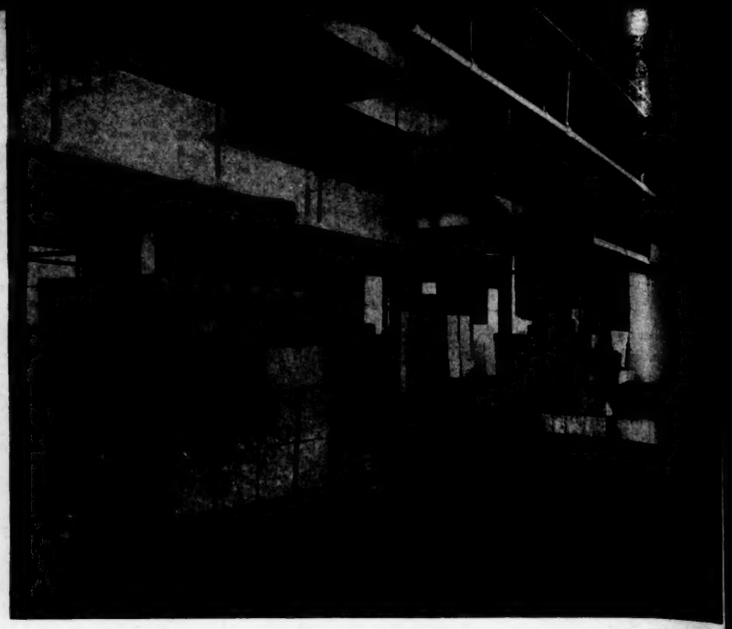


Fig. 14. In the storage of fine candies awaiting shipment, relative humidity can raise havoc. Dehumidifiers have been installed in this storage room to guard against any impairing of quality or appearance at this point.

satisfactory. A condition of 63F and 50% relative humidity is maintained.

In addition to those spaces already mentioned, an installation of a silica-gel dehumidifier in the starch room was accomplished more or less as an experiment. This worked out very well.

Conclusion

In conclusion it can be safely said that in the instance of the Lewis Candy Company the engineers and executives have recognized a fact that has long been in the minds of many men in charge of production—that there are two distinct and varying features in air conditions which seriously affect candy, both in production and in storage—temperature and humidity. These two are dependent somewhat, one upon the other, but are in no way tied one to the other. Because of this fact they must be treated as independent factors and cannot be satisfactorily disposed of by attempting to control either but not both.

Effect of Atmospheric Conditions on Humans

In a recent talk before the Air Conditioning Council of Western New York, Dr. Albert G. Young, Director of Corey Hill Hospital, Brookline, Massachusetts, pointed out some of the effects of temperature and humidity on human beings. In his talk he referred particularly to the effect of changes in atmospheric conditions on the heart and blood pressure. Abstracts from his paper are as follows:

The effect of temperature and humidity on the human body is of greater significance than has been previously supposed. The body, like an internal combustion engine, functions best at an optimum temperature. If this temperature varies too greatly in either direction, marked physiological changes result; thus, if the individual is exposed to severe cold, even over a small area of the body, the blood pressure rises. This is shown by immersing the hand in ice water—the blood pressure will increase in the normal individual as much as 30 to 40 millimeters mercury in the systolic pressure with a corresponding rise in the diastolic pressure. In the normal individual this increased pressure is maintained for only 30 to 60 minutes.

Any increase in body temperature or any attempt on the part of the body to dissipate its heat formation is marked by an increase in the circulation. This is due to the fact that the skin is the chief source of heat dissipation from the body. If the atmospheric temperature is increased, the normal body temperature can be maintained by diverting some of the blood through capillary circulation in the skin, which means that there will be less circulation to the internal organs. In a dry atmosphere a normal temperature can be maintained if the individual is able to perspire when the surrounding temperature is as much as 120F to 130F. Under a high humidity the body is unable to dissipate heat by evaporation and this is followed by a rise in temperature; the circulation to the skin is increased; the heart beats faster, and respiration is increased. This may not be sufficient to maintain a normal physiological condition. As the capillary circulation in the skin is increased, there is a fall in blood pressure. This fall is quite marked, enough to give the normal individual definite symptoms. The changes in heart and circulation probably explain the fact that under increased temperatures and humidities the normal individual fatigues more rapidly, with a consequent decrease in his ability to perform either physical or mental work.

DUST CONTROL

in the granite industry is discussed in this article, the third of a series which describes dust collectors and filtration.

By WESLEY C. L. HEMEON†

CLOTH filter dust collectors are very efficient in the cleaning of dusty air and, in the case of granite cutting sheds, are much lower in first cost than wet type collectors. They are, therefore, employed almost exclusively for this purpose.

Cloth dust collectors are of two general types—those in which the unit of cloth is mounted on a rectangular frame and those in which it is in the shape of a cloth tube suspended vertically, the units in both cases being closely spaced in any desired number to give a large area of cloth encompassed within a small volume.

Filters are defined as to size in terms of the free cloth area or filtering area. The greater the area of cloth in a given set of circumstances, the less is the pressure drop. It is obviously of importance to provide a filter of adequate size, but economic considerations require that it not be excessive. It is common practice to specify a filter of such size that the filtration velocity (air flow in c.f.m. divided by the cloth area in square feet) will be about 3 ft. per min. There has, heretofore, been no engineering basis on which to estimate the pressure drop to be expected through a filter in any given circumstance.

Hatch described[1] the importance of establishing the relationship between the pressure drop and various factors such as dust loading, dust fineness and degree of packing and has undertaken a comprehensive laboratory study of the problem. Such information would make it possible to compute the pressure drop for any given set of conditions or to select a filter of a size to result in a specified pressure drop. The author, stimulated by Hatch's suggestions, made a study of the operating characteristics of commercial size filters handling granite dust, the results of which are reported herein.

Theory of Friction in Filters

It is well known that the pressure drop through filters bears a first power relationship to the filtration velocity, proving that air flow through the pores of the cloth and of a layer of dust is in streamline motion as contrasted with the turbulent motion of air flow in duct work where resulting friction losses vary as the square of the velocity.

We can therefore express the pressure drop through the cloth of a dust filter in these terms:

$$R_o = K_o V_o \quad \ldots\ldots (1)$$

Where R_o is the pressure drop through the cloth, K_o the constant (basic cloth resistance) which can only be determined empirically, and V_o the filtration velocity.

The pressure drop through a layer of dust supported on filter cloth is likewise directly proportional to the filtration velocity. It is also proportional to the thickness of the dust layer (length of travel of the air stream) and may vary with the nature of packing and with the fineness of the dust (1). The thickness of the dust layer may be expressed, at least approximately, in terms of weight per unit area of cloth (dust loading), which is much more convenient to measure than thickness. If, further, we assume degree of packing and fineness of granite dust, as usually generated, to vary only within a narrow range, we can lump all factors, with the exception of dust loading, into a single proportionality constant and express the pressure drop across the dust layer as

$$R_1 = K_1 V_1 w \quad \ldots\ldots (2)$$

Where R_1 is the pressure drop across the dust layer, K_1 the constant (basic dust resistance) to be deter-

†Engineer, Division of Occupational Hygiene, Massachusetts Dept. of Labor and Industries.

[1]Private Communication, Nov. 1938.

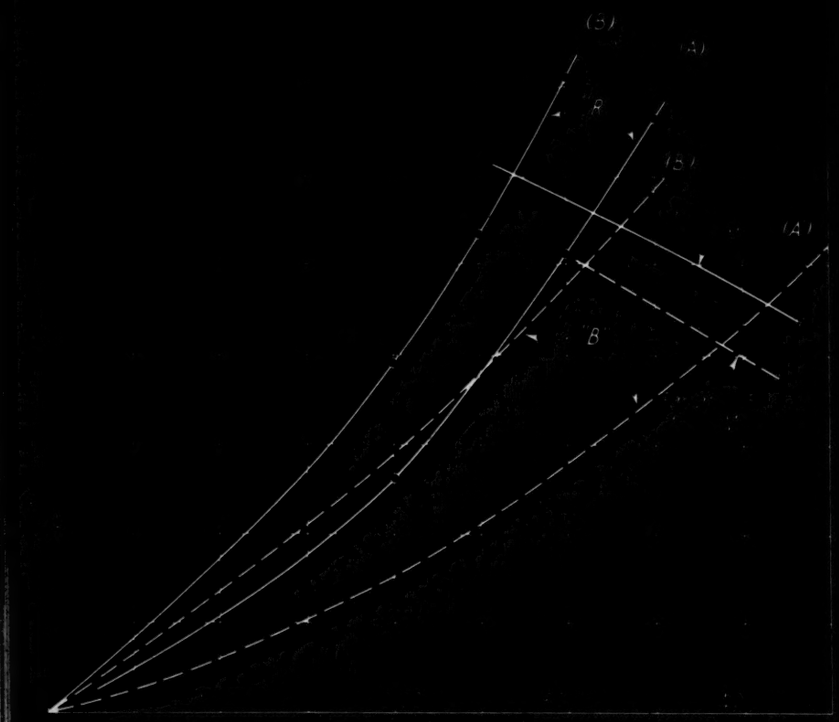


Fig. 15. Pressure air flow characteristics of an exhaust system including a filter, together with characteristics of a suitable fan. Intersections define flow conditions at beginning and at end of duct cycle.

mined experimentally, V_1 the filtration velocity through the loaded filter, and w the dust loading.

The pressure drop, R, across the loaded filter will be the sum of R_o and R_1. That is

$$R = K_o V_1 + K_1 V_1 w \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

In order to render the expression of equation (3) useful for the selection of economical filter sizes, it is necessary to be able to estimate the values of K_o and K_1, as well as the rate of dust production, the latter to fix the value of w.

Test Method

Data have been secured from several tests on four commercial filters operating in granite cutting sheds in conjunction with exhaust systems for hand pneumatic tools, surfacing machines and abrasive blasting. In addition, other confirmatory data not here reported have been obtained.

The test procedure follows. After an operating period of suitable duration, the static pressure on both dirty and clean sides of the filter was measured with a U-tube at a small hole in the wall of the collector to avoid the pressure losses due to entrance and exit connections, and the rate of air flow determined by measurements with a Pitot tube on a 6-point traverse in a connecting duct. The fan was then stopped and, before shaking the bags, the hopper was opened to remove therefrom any chips or *stone sand* that had settled out before reaching the cloth. After shaking the bags for two or three minutes, the dust was removed from the collector hopper, weighed, and the fan again started. Pressure drop and air flow measurements were then repeated.

The basic cloth resistance, K_o, is calculated from the values of pressure drop, R_o, in inches of water and filtration velocity, V_o, in feet per minute, obtained immediately after shaking (Equation 1). Then, by substitution in equation (3) of the calculated value of K_o and the dust loading, in ounces per square foot, together with values observed before shaking—pressure drop, R, in inches of water and filtration velocity, V_1, in feet per minute, one can calculate K_1. Employing these units, it will be noted that the basic cloth resistance, K_o, is the pressure drop through the cloth in inches of water at a filtration velocity of 1 foot per minute, and the basic dust resistance, K_1, is the pressure drop through the dust layer in inches of water at a filtration velocity of 1 foot per minute and a dust loading of 1 ounce per square foot.

The results of several tests on cloth tube collectors are summarized in Table 1.

Tests numbered 1 to 4 were conducted on the same filter over a period of a year. It will be noted that, with due allowance for experimental error, the maximum value for K_o is about 0.25. The higher value in test No. 3 is overshadowed by three other values on the same filter. The value reported in test No. 9 is believed due to the hand-operated shaking mechanism with which this filter was equipped. Although the same was true of the filter represented in tests 6-8, this was a newer filter and had received careful attention. It will not be surprising if, in the course of time, its cloth resistance also gradually increases. Hand-operated shakers for filters are, in general, quite inadequate.

The data on values of K_1 justify taking 0.20 as an average value. There is reason to believe that K_1 may become appreciably smaller at higher dust loadings but, for the present, calculations may be based on this constant in confidence that they are on the conservative side.

Several tests on flat-bag collectors handling stone dust similar to granite dust, together with one test on such a collector handling granite dust, lead to the conclusion that a good working figure for the basic cloth resistance of this type of filter is 0.85. (See Table 2).

TABLE 1.—RESULTS OF TESTS ON CLOTH TUBE DUST COLLECTORS

Test No.	Plant	Date	Filtration Velocity, Loaded	Filtration Velocity, Clean	Pressure Drop, Loaded	Pressure Drop, Clean	Dust Loading, w	Basic Cloth Resistance, K_o	Basic Dust Resistance, K_1
		Surfacing Machines							
1	W	11/21/38		2.35		0.50		0.22	
2	W	11/23/38	2.24	2.35	1.40	0.40	2.85	0.17	0.15
3	W	5/23/39	2.15	2.26	0.90	0.60	0.87	0.27	0.14
4	W	10/24/39	2.01	2.27	1.10	0.50	1.45	0.22	0.23
5	R	11/2/39	3.73	3.89	1.55	0.65	1.06	0.18	0.21
		Banker Exhaust (Hand Tools)							
6	Mc	8/15/39	7.58	7.60	2.20	1.35	0.55	0.18	0.20
7		9/8/39		8.20		1.40		0.17	
8		11/3/39	8.05	9.05	3.40	1.85	0.48	0.20	0.20
		Abrasive Blasting							
9	M	11/9/39		4.48	2.35	1.80	0.53	0.32	0.23

TABLE 2.—TYPICAL TEST DATA ON FLAT BAG COLLECTORS FOR DETERMINING BASIC CLOTH RESISTANCE

Cloth Area	Dust Handled	Filtration Velocity	Pressure Drop	Basic Cloth Resistance
2244 sq. ft.	Stone	1.59 f.p.m.	1.25"	0.79
484 sq. ft.	Stone	4.42 f.p.m.	3.65"	0.83
242 sq. ft.	Stone	4.79 f.p.m.	3.95"	0.83
2574* sq. ft.	Granite	2.76 f.p.m.	2.6"	0.94

*Older model collector.

In recapitulation, the following constants are recommended for use in calculating filter resistances for granite dust:

Basic cloth resistance, K_o, (cloth tube filter)..0.25
Basic cloth resistance, K_o, (screen type filter).0.85
Basic dust resistance, K_1, (granite cutting)....0.20

Rate of Dust Production

Data as to amounts of dust produced in each of the granite cutting operations is of importance not only for use in estimating pressure drops through dust filters, but also as an aid to a proper appreciation of the relative responsibility of each process for recontamination of working atmospheres and for neighborhood dust nuisance.

Weights of fine dust removed from cloth collectors (excluding coarse material that settles in the hopper), representing that produced in known periods of time, have been obtained and are set forth in Table 3.

The data for rate of dust production from surfacing machines given in Table 3 are for continuous operation, but the total daily quantity will, of course, be dependent on the number of hours the machine is in actual use. The same applies to the data on abrasive blasting operations.

The data of item 3, showing amounts of dust produced from hand pneumatic tools, are based upon daily weighings of dust from a 2-banker filter for 15 days, including various kinds of work. The data of item 4 cover hand tool work on building stone with working air pressures of 100 lbs. per square inch. This figure can be said to represent the maximum dust production from hand-held tools.

A consideration of the data of Table 3, together with the circumstances of each test and certain supplementary data not tabulated, lead to the figures set forth in Table 4. These are recommended for use in estimating sizes of filters and pressure drops through them.

Estimating Filter Sizes and Pressure Drops

Sufficient data have been presented to permit close estimates of necessary cloth area in a filter for any fixed pressure drop. The method may be illustrated by a few practical examples.

Example 1. A filter of the cloth tube type is desired for the exhaust of a surfacing machine, with a minimum air flow of 600 c.f.m. It is decided to allow for a pressure drop through the filter of about 5" at the end of cycle (just before shaking). Working periods are 7:30 A.M. to 12:00 and 1:00 to 4:30 P.M. The schedule will call for shaking the bags each noon and at the end of the day.

Solution: Taking the longer morning period of 4½ hours, we assume that there will be not over 4 hours intensive dust production. Reference to Table 4 indicates that a dust production rate of 20 lbs. per hour will not be exceeded. Therefore we calculate 80 lbs. of dust on the cloth at the end of the dust cycle.

The basic cloth resistance of the cloth tube collector is 0.25, the basic dust resistance 0.20.

A cut and try method of solution, using equation (3), is simplest. As a first try, let us assume a filter with 200 sq. ft. of cloth.

Filtration velocity, $V = 3$ f.p.m.

$$\text{Dust loading, } w = \frac{80 \times 16}{200} = 6.4 \text{ oz. per sq. ft.}$$

Substituting in the equation

$$R = K_o V + K_1 V w$$
$$= 0.25\ (3) + 0.20\ (3)\ 6.4$$
$$= 4.6 \text{ in.}$$

Which is sufficiently close to the maximum pressure drop specified and we therefore conclude that a filtration velocity of 3 f.p.m. satisfies the stated conditions.

Example 2. Using the other data of Example 1, assume instead that a minimum air flow of 1200 c.f.m. is desired. Calculate the pressure drop through a cloth tube filter of a size to result in a filtration velocity of 4 f.p.m.

Solution:

Filtration velocity = 4 f.p.m.

$$\text{Cloth area } \frac{1200}{4} = 300 \text{ sq. ft.}$$

$$\text{Dust loading} = \frac{80 \times 16}{300} = 4.25 \text{ oz./sq. ft.}$$

Substituting in equation (3)

$$R = 0.25\ (4) + 0.20\ (4)\ 4.25$$
$$= 4.4 \text{ in.}$$

Example 3. A cloth tube filter is to be used in conjunction with an exhaust system for 6 bankers. A 5-hour work cycle (before shaking the filter) is to be

provided for and the air flow at the end of the cycle must not be less than 400 c.f.m. through any exhaust inlet. It is established that this means a total minimum air flow of 2700 c.f.m. Calculate the pressure drop at the end of the cycle for a filtration velocity of 10 f.p.m.

SOLUTION: We note by reference to Table 4 that the maximum amount of dust to be expected in this situation, employing hand pneumatic tools, is 1 lb. per hour per man, that is, in 5 hours an aggregate of 30 lbs.

$$\text{Cloth area} = \frac{2700}{10} = 270 \text{ sq. ft.}$$

$$\text{Dust loading, } w = \frac{30 \times 16}{270} = 1.78 \text{ oz./sq. ft.}$$

Substitution in equation (3) gives

$$R = 0.25(10) + 0.20(10)\ 1.78$$
$$= 6.1 \text{ in.}$$

Example 4. It is desired to construct a single exhaust system to take care of a surfacing machine, 6 banker exhaust units and an abrasive blasting room, with a single filter to be shaken on a 4-hour schedule. Select a filter size to result in a maximum pressure drop of about 5 in. at the end of 4 hours.

DATA

	Minimum Air Flow	Pounds Dust
1 surfacing machine	600 c.f.m.	80
6 banker exhaust units	2700 c.f.m.	24
1 abrasive blasting room	1000 c.f.m.	28
Totals	4300 c.f.m.	132 lbs. dust

SOLUTION:

Try filtration velocity, V, = 6 f.p.m.

$$\text{Area} = \frac{4300}{6} = 717 \text{ sq. ft.}$$

$$\text{Dust loading, } w, = \frac{132 \times 16}{717} = 2.96 \text{ oz./sq. ft.}$$

$$\text{Then } R = 0.25(6) + 0.20(6)\ 2.96$$
$$= 1.5 + 3.55 = 5.1 \text{ in.}$$

We conclude that a 6 f.p.m. filtration velocity results in the specified maximum pressure drop.

Variation in Air Flow Caused by Filter

The total resistance of the system will vary continuously throughout the cycle due to the changing pressure drop through the filter as it accumulates dust. The extent of this variation and the consequent variation in air flow can only be estimated by comparing the static-pressure-volume characteristics of the system with those of the fan to be used, and this may be done simply by plotting these two on the same graph. An analysis of Example 3 by this method follows.

Using flexible unit R (see Fig. 1, Part 1) it is estimated (16) that with 6 bankers the static pressure at the fan at an air flow of 2700 c.f.m. due to duct resistance alone (in the absence of a filter) is 6 in. Hence, $Q = 1100\sqrt{R_d}$, where Q is the air flow in cubic feet per minute (2700) and R_d is the resistance of the duct system (6 in.) not including the filter. Calculation of the total duct resistance and filter resistance for several selected rates of flow gives the total system resistance. This is done for the clean filter just after shaking and again at the end of the cycle with the filter loaded as indicated in the following tabulation.

The figures in the first and last columns are plotted in Fig. 15 as curves (A) and (B) together with the characteristics of a commercial steel plate exhaust fan. The dotted lines (A′) and (B′) represent the characteristics of the same exhaust system using flexible device B (see Fig. 2, Part 1) whose lower resistance results in an estimated (16) total duct resistance of 3 in. at the fan (not including filter). A slightly greater air volume of 3100 c.f.m. is provided to care for unbalanced

TABLE 3.—OBSERVATIONS ON AMOUNTS OF DUST PRODUCED IN VARIOUS GRANITE CUTTING OPERATIONS

No.	Operation	Conditions	Pounds Dust per Hour per Man
1	Surfacing machine	Continuous operation, 3 hrs., with 4-point chisel—80 p.s.i. air pressure	19.6
2		Continuous operation, 2¾ hrs., with 4-point chisel—80 p.s.i. air pressure	17.0
3	Hand pneumatic tools	Discontinuous operations of 2 men with miscellaneous chisels on small monuments 8 hrs. per day. Based on weights on each of 15 days. 80 p.s.i. air pressure Minimum	0.20
		Maximum	0.80
		Average	0.35
4		Discontinuous operations of 14 men with miscellaneous chisels on building stone. Based on 2½ days operations, 280 man hours. 100 p.s.i. air pressure	0.5
5		Similar to above. Discontinuous operations of 15 men except abnormal amount of cutting with plug drills 2 days	1.8
6	Abrasive blasting	Continuous operation, 3 hrs., carving letters and ornamentation on face of monument. Artificial abrasive	4.5
7		Continuous operation, 1½ hrs., carving letters and ornamentation on face of monument. Artificial abrasive	7.3

TABLE 4

MAXIMUM RATES OF DUST PRODUCTION FROM VARIOUS GRANITE CUTTING OPERATIONS FOR USE IN ESTIMATING PRESSURE DROPS THROUGH FILTERS AND RELATIVE IMPORTANCE OF EACH IN CONTRIBUTING TO NEIGHBORHOOD DUST NUISANCE

Operation	Dust Production Pounds per Hour per Man
Hand pneumatic tools on granite monuments	½ to 1
Hand pneumatic tools on granite building stone	½ to 2
*Heavy surfacing machines—4-point chisel	15 to 20
Abrasive blasting of lettering, etc.	4 to 7

*No data have been secured as to dust production from so-called "medium" size surfacers, nor from bush hammering, but both produce very much less than the 4-point chisel in a heavy surfacer. An estimated value of 3 lbs. per hour is suggested for bush hammering and, for medium surfacers, 8 lbs. per hour.

(A) At Beginning of Cycle

Air Flow	Duct Resistance $R_d = \left(\frac{Q}{1100}\right)^2$	Filter Resistance $R_o = K_o\left(\frac{Q}{A}\right) = 0.25\left(\frac{Q}{270}\right)$	Total System Resistance
2000 c.f.m.	3.3 in.	1.85 in.	5.2 in.
3000 c.f.m.	7.4 in.	2.78 in.	10.2 in.
3500 c.f.m.	10.1 in.	3.25 in.	13.3 in.

(B) At End of 5 Hrs. Cycle

Air Flow	Duct Resistance $R = \left(\frac{Q}{1100}\right)^2$	Filter Resistance $R = K_o\left(\frac{Q}{A}\right) + K_1\left(\frac{Q}{A}\right)^w = 0.25\left(\frac{Q}{270}\right) + 0.20\left(\frac{Q}{270}\right)^{1.78}$	Total System Resistance
2000 c.f.m.	3.3 in.	4.5 in.	7.8 in.
2500 c.f.m.	5.1 in.	5.6 in.	10.7 in.
3000 c.f.m.	7.4 in.	6.75 in.	14.2 in.

flow wherein hoods nearest the fan draw more air than those more remote. (The same cloth area, 270 sq. ft., has been employed, resulting in a higher filtration velocity.)

The intersection of these system characteristics with the fan characteristics define flow conditions with respect to air volume and static pressure at the beginning and at the end of the cycle, from which power requirement may be calculated. The fan characteristic curves plotted were, of course, selected at speeds to result in intersection with the (B) curves at the appropriate points. Calculations of power requirement, assuming a 60% fan efficiency, are included in the following recapitulation and tabulation of results.

		Start of Cycle	End of Cycle
Device R	Air Flow	3150 c.f.m.	2700 c.f.m.
	Static Pressure at Fan	11.2 in.	12 in.
	Power Required	9.3 H.P.	8.5 H.P.
	Filtration Velocity	11.6 f.p.m.	10 f.p.m.
Device B	Air Flow	3900 c.f.m.	3100 c.f.m.
	Static Pressure at Fan	8.3 in.	10 in.
	Power Required	8.5 H.P.	8.1 H.P.
	Filtration Velocity	14.4 f.p.m.	11.5 f.p.m.

It will be noted that there is no intolerable variation in either air flow or in power required from beginning to end of the cycle. Serious fluctuations in air flow could occur if the fan selected were of improper size or type and were caused to operate too near the peak of its static-pressure-volume curve. Proper care in examination of manufacturer's tables will avoid this error.

Allowable Pressure Drop

No blanket rule can be offered as to the pressure drop to be allowed through filters. The higher the drop, the greater the power required for a fixed air flow. On the other hand, larger filters because of greater initial cost are responsible for higher fixed charges which should be balanced against the lower operating cost in arriving at a decision as to the size of filter desired.

A cost analysis of this sort is set forth in Table 5 for the conditions described in Example 3, showing the power requirement and operating cost for cloth tube filters of various sizes as balanced against fixed charges. The figures are intended to be illustrative only. In the conditions specified in that analysis the minimum total operating cost is seen to be for a filter having around 500-900 square feet of cloth. Obviously, where a plant does not use its equipment as much as 150 days per annum, a smaller size filter might be justified. Variations in first cost from those set forth in the Table will likewise shift the optimum. Special financial considerations may enter a given case and dictate selection of a smaller filter in spite of greater operating cost.

Care should be taken to avoid locating exhaust inlets that lead to a filter, close to any wet process that might result in condensation of moisture inside the dust collector. This may happen in winter weather when the filter is located outdoors. If this situation is unavoidable, then the filter should either be located indoors (preferably with hopper discharge spouts leading to the exterior) or else the collector housing insulated as by constructing a room around it that communicates with the heated workroom. In this case the hopper discharge spout should pass through the floor of such a chamber to the exterior.

Filters of the cloth tube type do not require more than 3 to 4 minutes of shaking to free the bags of adhering dust. The practice of shaking for 10 minutes or more is not recommended, as it constitutes an unnecessary wear and tear on the cloth.

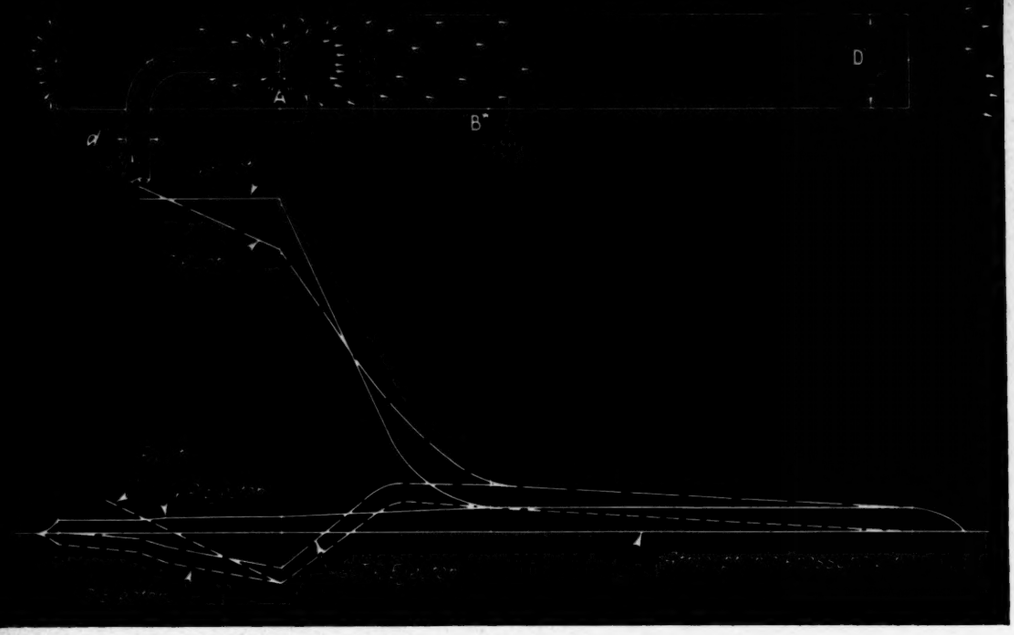
If these precautions are observed, together with the elementary ones of shaking bags at least twice a day and emptying hoppers before they are full, there appears to be no good mechanical reason why filters of the cloth tube type should always be limited to the sizes commonly fixed by the 3 f.p.m. rule.

The various considerations outlined above provide a rational basis for selecting sizes to fit individual conditions. Although the figures described and specific conclusions reached apply only to the dust resulting from granite cutting, the principles are applicable to the filtration from air of all kinds of dust.

TABLE 5.—ECONOMIC BALANCE APPLIED TO A BANKER EXHAUST FILTER (CLOTH TUBE TYPE) UNDER STATED CONDITIONS

Total Air Flow	2700 c.f.m.							
Filtration Velocity, f.p.m.	10		5		3		1½	
Filter Area, sq. ft.	270		540		900		1800	
Pressure Drop, Water Gage	6.1 in.		2.1 in.		1.1 in.		0.45 in.	
Horsepower	4.32		1.49		0.78		0.32	
Annual Power Cost*		$158		$54		$28		$11
Fixed Charges								
Original Cost	$240		$330		$425		$690	
Interest, 6%								
Depreciation, 10%								
Maintenance, 5%								
Total Fixed Charges, 21%	$50	$50	$69	$69	$89	$89	$145	$145
Total Annual Cost		$208		$123		$117		$156

*Computed for 150 working days annually, and a power cost of 4c per Kw.-Hr.



14 A straight tube ejector in which considerable energy is lost due to eddy currents produced in the mixing chamber. Lower section shows pressure conditions in the ejector. Note at "A" the static pressure is negative which causes the air to flow inward. As the resistance at the ejector discharge is increased, the static pressure at "A" also increases until it becomes positive when air at the induction end will back out rather than be drawn in.

The Venturi Ejector for Handling Air

PART 2—AN ANALYSIS AND PERFORMANCE FORMULAE

By F. F. KRAVATH†

THE ejector has been, and can be explained in many ways, all of which are, probably, correct. The accepted explanation credits the jet air, at its emergence into the mixing chamber of the ejector, with impacting the still air directly in front of it and by the very nature of its velocity and mass, i.e. its momentum, forcing this air all the way out of the tube. The air directly behind the impacted air has now become rarefied, and hence lighter, allowing the heavier atmospheric air to push its way into the tube at the induction end and advance to the mixing chamber. Fig. 14 illustrates the emergence of jet air into a straight tube ejector (which oddly enough is not much less efficient than the venturi ejector) and indicates the turbulence which exists in the mixing chamber. Considerable energy is lost here in the formation of eddy currents, which are not entirely dissipated until the air has reached a point from 7 to 9 diameters from the mixing chamber. By converging the induced air stream, the velocity of the induced air is accelerated so that the difference in velocity between the jet and the induced, the primary and secondary air is at a minimum, and hence the shock loss and eddy current formation is also a minimum. Fig. 15 shows the familiar venturi shape, with the converging cone for more efficient mixing, the straight section at the throat for a gradual straightening of the air stream preparatory to regain of static or potential energy again in the diverging section. Without the straight section, a vena contracta loss would occur a short distance beyond the narrowest section. One reason why the straight tube ejector is not as inefficient as it seems (by comparison with the venturi ejector) is that while it is inefficient from a mixing angle, it offers less resistance to the flow of the mixed air from the mixing chamber to the discharge. The first consideration, however, far outweighs the second.

The writer, in any explanation of the ejector principle, likes to stress the fact that it is because of conversion of energies and manipulation of pressures that the ejector works. The total pressure of the jet air stream at its emergence into the mixing chamber must be less than the velocity pressure at this point, otherwise no induction will occur. The greater the difference between the two pressures, for any fixed mechanical resistance, the more air will be induced. This is all pictured graphically in Fig. 14. Here, the pressure conditions existing at every point along the ejector are set forth and the manner in which they tie together becomes apparent. Starting with the jet velocity pressure which is always high at the jet discharge, we proceed through the mixing chamber where the velocity and velocity pressure fall rapidly until the point "B" is reached at which the jet and induced air velocities have become stabilized. From this point until the ejector discharge, the velocity pressure remains constant. At the discharge, it decreases again until, some distance beyond, it has reached zero. Referring, now, to the induction end of the tube, we notice that some distance in front of the tube entrance the velocity pressure increases from zero to a value which it holds until the entrance of the jet into the ejector tube, at which point it is increased slightly. The velocity pressure now remains constant until the point "A" is reached at which it begins to increase slightly under the action of the

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jet stream until the point "B" is reached at which point the velocity pressure of the jet stream and the induced air stream are the same. Mixing is now complete. The increase in velocity pressure at the point where the jet enters the ejector is due to the decrease in the effective, air-carrying cross-sectional area of the duct, and the corresponding increase in velocity.

Examining the total pressure curve of the jet, we notice that it decreases from a maximum value at the extreme left to a value somewhat less than the velocity pressure at "A." It is the difference between the velocity and total pressures at this point, or the static pressure, which, being negative, causes air to flow into the tube as will shortly be explained. Continuing, however, with the total pressure curve, it decreases sharply in value from "A" to "B" due to the extreme shock encountered by the high velocity jet stream. This loss is represented by the familiar Borda's equation as being the pressure corresponding to the difference in velocities between the jet and the mixed air. Continuing, the frictional resistance of the pipe is encountered, and finally at the ejector discharge, the complete velocity pressure loss at this point. Again returning to the induction entrance, the total pressure at this point is, of course, zero indicating no work done, as yet. As we proceed further inward, we encounter the orifice loss at entry (which is extremely slight), the impact loss at the jet entry, and various pipe resistances up to the point "A." The question might be asked, "Why is the total pressure shown negative instead of positive up to this point?" Since flow is inwards by virtue of the negative value of the static pressure at "A," which is the same for both the jet and the induced air, as well as for every other point of the pipe at "A," it is obvious that the total pressure which is always the algebraic sum of the static and velocity pressures, will be negative for the induced air section of the tube. As we move beyond "A," work is done on the induced air, and the total pressure curve, as well as the static pressure curve, changes its direction. Since the rarefaction of the air is destroyed at impact and it is compressed, we can understand the static pressure curve changing from a negative to a positive value, all within the mixing chamber. The total pressure curve is similar to the TP curve for a fan both exhausting and blowing. Within the fan, where work is done on the air, the TP curve changes from its maximum negative value to its maximum positive. Continuing, since at "B" stable conditions are assumed to have been reached within the ejector, the TP curves of the jet and induced air streams coincide, and from that point to discharge are one and the same.

Since it is static pressure which causes flow, and since the value and type of this pressure (positive or negative) determines how much air will flow and in which direction, it is important that the next paragraph be understood, for it contains the explanation of the ejector.

At the point "A," the velocity pressure of the jet exceeds the total pressure; hence, there exists at this point a negative static pressure which causes air to flow inwards. The total pressure is less than the velocity pressure simply because the tube resistance from "A" to discharge is less, in pressure units, than the velocity pressure. If we increase this resistance, TP at "A" will become greater, SP will become smaller, and less air will be induced. If enough resistance is added so that TP exceeds VP at "A," SP will be positive, and air will blow out of the induction end. That is the reason the resistance at discharge of the ejector should be kept at the absolute minimum.

Fig. 15 illustrates the venturi ejector and its pressure graph. Labeled *primary* are the velocity pressure, static pressure and total pressure curves of the jet air, while the pressure curves of the induced air are labeled *secondary*. It can be seen that fundamentally, the same conditions result, except that the shock loss due to the jet air piling up on the induced air has been reduced somewhat, as has the loss due to discontinuate flow in the mixing chamber.

Since the theory has been explained, let us see if we can derive a formula for the ejector which will tie together the various important elements, so that we may be able to predict with reasonable accuracy the performance which may be expected for any set of design conditions.

In Fig. 16 is illustrated the effect of the ratio of the Venturi Ejector discharge diameter to the Jet discharge diameter on the constant K_2. This constant is a function of the velocity pressure in the discharge of the ejector, labeled "D" in Figs. 14 and 15, while the product of the two represents the pressure loss (in the TP curve) encountered in the Venturi Ejector from shortly

15 **A Venturi ejector with a converging cone for more efficient mixing and a straight section at the throat for a gradual straightening of the air stream. Note that pressure conditions are much the same as on the straight tube ejector except that the shock losses due to the jet air piling up on the induced air has been reduced somewhat.**

beyond the jet discharge until just preceding the ejector discharge. These values of K_2 were arrived at experimentally, and are to be used only for insertion in the performance formula of the Venturi Ejector and not the Straight Tube Ejector.

Also of interest is the characteristic shape of the jet, as shown in Fig. 15. The jet pipe diameter is always kept large so that the velocity will be low and the corresponding pipe friction loss also low. It is only just before discharge into the ejector that the air stream is converged through a 15-degree cone so as to deliver the proper jet discharge velocity.

Performance Formula for Venturi Ejector

The derivation of this formula follows:

At any point in any duct

$$TP = SP + VP \quad \ldots\ldots(1)$$

where

TP = Total Pressure, in inches W.G.,
SP = Static Pressure, in inches W.G.,
VP = Velocity Pressure, in inches W.G.
V_d = velocity at jet discharge, in fpm,
V_D = velocity at tube discharge, in fpm,
K_3 = frictional factor in ejector entry up to point "A",
A_D = cross sectional area of ejector discharge, in sq. ft.,

CFM at inlet = volume at air induced into ejector

CFM at jet = volume of air through jet, tube, cfm.

hence,

$$SP = TP - VP \quad \ldots\ldots(2)$$

Referring to Fig. 14, at the point "A" in the ejector, the static pressure of the ejector tube and of the jet are equal. But TP at point "A" is made up of

$$\left[\begin{matrix}\text{the shock loss}\\ \text{at jet discharge}\end{matrix}\right] + \left[\begin{matrix}\text{frictional resistance of tube}\\ \text{from "A" to ejector discharge}\end{matrix}\right] + \left[\begin{matrix}\text{velocity loss at}\\ \text{ejector discharge}\end{matrix}\right]$$

Hence, TP (of Jet at "A") =

$$\left(\frac{V_d - V_D}{4006}\right)^2 + K_1\left(\frac{V_D}{4006}\right)^2 + \left(\frac{V_D}{4006}\right)^2 \quad \ldots\ldots(3)$$

and, SP (of Jet at "A")

$$= \left(\frac{V_d - V_D}{4006}\right)^2 + K_1\left(\frac{V_D}{4006}\right)^2 + \left(\frac{V_D}{4006}\right)^2 - \left(\frac{V_d}{4006}\right)^2 \quad (4)$$

$$= \frac{V_d^2 - 2V_dV_D + V_D^2 + K_1V_D^2 + V_D^2 - V_d^2}{(4006)^2}$$

$$= \frac{(2 + K_1)\,V_D^2 - 2V_dV_D}{(4006)^2}$$

$$= \frac{V_D[V_D(2 + K) - 2V_d]}{(4006)^2} = V_D\frac{[V_D \times K_2 - 2V_d]}{(4006)^2} \quad \ldots\ldots(5)$$

but,

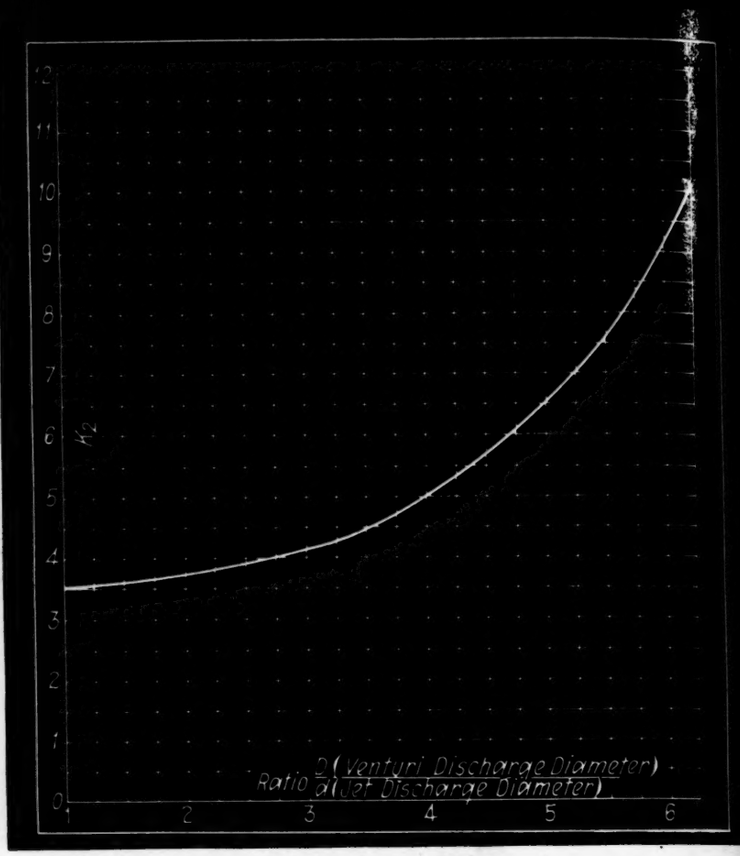
$$\text{CFM at discharge} = \text{CFM at inlet} + \text{CFM at jet} \quad \ldots\ldots(6)$$

or

$$A_DV_D = \text{CFM at inlet} + \text{CFM at jet}$$

and

$$V_D = \frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D} \quad \ldots\ldots(7)$$



16 **The effect of the ratio of the ejector discharge diameter to the jet discharge diameter on the constant K_2. This constant is a function of the velocity pressure in the discharge of the ejector. These velocities of the K_2 were arrived at experimentally and are to be used only for insertion of the performance formula.**

Substituting in (5),

$$SP \text{ (of Jet at "A")} = \frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D} \times \frac{\left(\frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D}\right)K_2 - 2V_d}{(4006)^2} \quad \ldots\ldots(8)$$

but,

$$SP \text{ (ejector at "A")} = -K_3\left(\frac{V \text{ at inlet}}{4006}\right)^2 \quad \ldots\ldots(9)$$

hence,

$$-K_3\left(\frac{V \text{ at inlet}}{4006}\right)^2 = \left(\frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D}\right) \times \frac{\left(\frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D}\right)K_2 - 2V_d}{(4006)^2} \quad \ldots(10)$$

and,

$$-K_3\,(\text{CFM at inlet})^2 = (\text{CFM at inlet} + \text{CFM at jet})\,[(\text{CFM at inlet} + \text{CFM at jet})\,K_2 - 2V_dA_D] \quad \ldots(11)$$

Example:—Assuming that 5325 c.f.m. are to be exhausted from a paint spray booth, and it is decided to use a venturi ejector of the shape of Fig. 2, but with a diameter at both ends of 36 in. instead of 12 in., the jet diameter is to be 1/6 of the duct diameter or 6 in.

REQUIRED:—to find the volume of jet air, and the jet air horsepower if the static pressure necessary to overcome all resistance up to the mixing chamber of the ejector is 0.18 in. WG (according to the mechanical resistance imposed and the velocity to be maintained).

From Formula (8)

$$SP = \frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D} \times \frac{\left(\frac{\text{CFM at inlet} + \text{CFM at jet}}{A_D}\right)K_2 - 2V_d}{(4006)^2}$$

Substituting in (8) the value of K_2 for $\frac{D}{d} = 6$, which from Fig. 13 = 9.2; for A_D the value of 7.1 sq. ft. which is the area of a 36-in. dia. circle and for A_d the value of 0.1964 sq. ft. which is the area of a 6-in. dia. circle, we get

$$-0.18 = \left(\frac{5325 + \text{CFM jet}}{7.1}\right) \times \frac{\left(\frac{5325 + \text{CFM jet}}{7.1}\right) \times 9.2 - \frac{2 \times \text{CFM jet}}{0.1964}}{(4006)^2}$$

$$-0.18 \times (4006)^2 = (755 + 0.14\ \text{CFM jet})(6930 + 1.3\ \text{CFM}_d - 10.2\ \text{CFM jet})$$

$$-2{,}900{,}000 = (755 + 0.14\ \text{CFM jet})(6930 - 8.9\ \text{CFM jet})$$

$$-2{,}900{,}000 = 5{,}240{,}000 - 5750\ \text{CFM jet} - 1.248\ (\text{CFM jet})^2$$

$$0 = 8{,}140{,}000 - 5750\ \text{CFM jet} - 1.248\ (\text{CFM jet})^2$$

Solving, we get

$$\text{CFM jet} = 1130$$

Therefore,

$$V_d = \frac{1130}{0.1964} = 5740\ \text{fpm.}$$

$$VP_d = \left(\frac{5740}{4006}\right)^2 = 2.06''\ \text{W.G.}$$

$$\text{Air horsepower} = \frac{1130 \times 2.06}{6356} = 0.365$$

Practical Application and Worth of the Venturi Ejector

For certain types of installations where, perhaps, the material being handled is extremely corrosive, explosive, hot, or sticky it is better practice to avoid running this material through a fan regardless of whether it is of the centrifugal or disc type. The venturi ejector offers the safer, more practical, method of meeting these problems. It is true that in corrosive gas or fume handling the fan and wheel may be painted so as to resist chemical action, but this is an expensive procedure if done correctly. Where hot gases are handled, bearings are often water cooled if fans are used, but this procedure, again, is costly. The ejector avoids corrosion to fan, explosions due to sparks and heat, warpage or burned-out bearings, and out-of-balance wheels by the simple expedient of handling none of the aforementioned critical material through the fan. Only atmospheric or room air goes through the fan.

Where it is possible to keep the conveying velocities reasonably low so that the overall resistance may not exceed one to one and one-half inches water gauge, abrasive materials which are exhausted to the atmosphere without filtering, may be profitably handled with the ejector, thus avoiding wear on the fan. Where noncorrosive fumes, or fine dusts are handled and where the nuisance or hazards are created only infrequently such as less than 25% of the day, or where the total volume of air to be exhausted is relatively small, use of the venturi ejector may represent a worthwhile saving in initial cost of equipment. Where the ejector is to compete with fans, extremely prudent selection of the blower, and its operating speed, as well as careful design of the blowing duct must be made.

The ejector, of course can be made to work, and has been, using steam, water, or exhaust gases under pressure as the inspirating medium. However, the use of steam is extremely wasteful inasmuch as practically all of the heat energy is wasted, and steam is expensive to make. Water, while not nearly as expensive as steam, nevertheless is more expensive than air. Again, it is heavier, and full utilization cannot be made of its greater mass, entailing a loss in operating efficiency. Where exhaust gases are available under a few inches of water gauge, an efficient ejector set-up may well be made.

There are many adaptations of the venturi principle, whereby suctions are produced in pipes otherwise under pressure. The principle, however, has been explained in detail in an article titled "Duct Pressures" by the writer which appeared in the February issue of HEATING & VENTILATING, and in any event differs but little from the ejector. The greatest difference lies in the fact that the *suction producers* are used more for that purpose than for the removal of any appreciable volume or amount of material.

The writer hopes that others will come forward with original material and supplement this work, for there is much yet to be known about the ejector.

Simple Ventilation System for Alabama Tunnel

A simplified system of ventilation is provided for the new $4 million vehicular tunnel under the Mobile River at Mobile, Ala., which will be completed and opened to traffic early in the Fall. This was made possible by the comparatively short length of the closed tunnel—about 3400 feet.

A feature of the tunnel is the running of the ducts under the roadway, although within the steel and concrete structure of the tunnel structure. Studies by the U. S. Bureau of Mines showed that although carbon monoxide rises when hot, it tends to fall when it cools and will fall if there is a downward movement of air as is the case in this tunnel where traffic moves in both directions through the same tube. Hence, in this installation the polluted air flows down the inclines and is drawn into the exhaust ducts through ports placed in the curb.

The ventilating building is placed on Blakely Island near one end of the tunnel. Air is exhausted from the tunnel over a length of about 400 feet, 200 feet on each side of the midpoint between tunnel entrances. Fresh air is drawn in through the tunnel entrances.

The fan equipment in the ventilating building is connected with the electric power supply at both ends of the tunnel. In addition a gas engine generating set in this building will provide power for lights, pumps and ventilation should both outside sources of service be interrupted by storm or other causes.

AIR BORNE DUST

Of the two articles which follow, the first is concerned with the elimination at the source of dust from the combustion of fuels, especially solid fuels. The second article discusses the removal of dust from air by electrical precipitation. Mr. Lincoln's article is abstracted from a paper presented before a meeting of the Massachusetts Chapter of the American Society of Heating and Ventilating Engineers while Mr. McConnell's paper is abstracted from an address before the Air Conditioning Bureau of Boston.

Dust and Fly Ash Removal from Boilers

By ROLAND LINCOLN†

WITHIN the past few years there has been a great increase in interest, not only in the air conditioning phase of clean air, but also in the elimination of dust at its source.

The burning of any fuel produces a combustion gas containing solid particles which are normally discharged at the chimney. The gas is diffused, but the solid particles either remain in the air or settle around us. If this solid material could be removed while still highly concentrated and under control in the duct work, everyone would benefit. It is about five thousand times as concentrated as it would be if diffused in the atmosphere. (2 grains per cu. ft. of flue gas vs. 4 grains per 10,000 cu. ft. of city air.)

Our problem consists in separating the solid particles from the combustion gas and we must know about these particles. They are called "Aerosols" (or solids in air) and may be divided into smokes, fumes and dusts. Technically, smoke is unburned carbon in extremely small sizes—so small that they can hardly be seen even with a microscope. Particles from 0.1 to 0.3 microns across are called smoke. While not visible individually, in dense concentrations they can be seen for miles. The production of smoke is due to improperly designed equipment or incorrect burning. Its elimination is up to the boiler manufacturer and the operating engineer, not to the dust collector.

Soot consists of flakes of unburned carbon produced by deposit on a cold surface. It settles rapidly and is a very serious offender as a public nuisance. Cinders are small bits of molten ash and carbon blown into the gas. They are usually formed in grate type boilers, particularly at high combustion rates, and settle rapidly, both in the duct work and outside the plant. Abrasive action is severe, especially at high fan speeds. Their nuisance value is high, but they are readily collected in most dust collectors. Fly ash is the name for ash particles carried into the air stream. Pulverized fuel plants, because the combustion takes place in the air, discharge from half to three-quarters of their ash with the flue gas. Pulverized fuel is a primary reason for the present interest and activity in the power plant dust collecting field. *Flue dust* is the term recommended in the new ASME Dust Collector Code. Flue dust includes all solid particles in flue gas larger than one micron.

A pulverized fired flue dust will have a very low carbon content (possibly 10%) with little change in carbon content due to the rate of burning. A stoker fed flue dust, however, will be very materially affected by the rate of combustion, and will vary from approximately one-fourth carbon to three-fourths carbon as the combustion is increased.

The ash will usually be in the form of glass-like fused spheres, with occasional bits of ash unaffected by temperature. With stoker firing, carbon will be embedded in these spheres, producing the well known cinder. In pulverized firing, the coal is in extremely small sizes and there is little opportunity to combine before burning; and for this reason the spheres will be almost all ash, and much smaller. Free carbon in the form of soot will occur where too low a temperature exists. Often the high carbon flue dust, if in large sizes, can be reintroduced to the furnace and so burned. It should be fed to a high temperature region well mixed with plenty of air.

Variations in the carbon content introduce another variable which affects the performance of dust collectors—that is, specific gravity. Flue dust will vary in specific gravity from 1.5 to 2.5 with an average of 2.0. Size and variation in size has an enormous effect on collection efficiency. Dust is measured in microns, a micron being 1/1000th of a millimeter, or 1/25400 part of an inch. Information is available on particles down to 1/1000 of a micron. Percentage of dust in samples found in each division (by microns) is termed the size distribution and may be tabulated, plotted, or otherwise recorded.

Performance of dust collectors is affected by the size of the dust particles. In general, centrifugal and inertia types of collectors have their highest collection efficiency for the large dust particles. The actual quantity of flue dust in the combustion gas depends upon the quantity of coal burned, its ash content and upon the method of burning. With underfed stoker firing, up to one-third of the ash with an equal quantity of carbon will be in the gas, depending upon the combustion rate. Spreader stokers will have about half the ash with a smaller percentage of carbon, while a pulverized fired

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boiler may have two-thirds of the ash in the gas, but comparatively little of the carbon. In general the dust loading will be between one and four grains of flue dust per cubic foot of gas.

An average dust loading of two grains per cubic foot is usually used.

Distribution of the different sizes affects the collector installation in many ways. Erosion of the induced draft fan is affected by the quantity of dust passing through it, particularly of the large sizes. Under severe conditions, when the induced draft fan must operate against a high pressure, the fans may be cut through in a few months if no collectors are installed. This is a distinct problem in the large central stations and especially so when using pulverized fired coal at high rates. Often the saving in the induced draft maintenance will more than pay for the dust collector installation. Reliability and availability of the boiler will be increased.

Size of the collector influences the efficiency somewhat, although not as much as some people claim. A reduction in velocity will reduce the efficiency, although not as much as sometimes claimed. Air leakage into the dust hopper will materially reduce the efficiency and is prevented either by using a double check valve, a rotary feed valve, or continuous ejector discharge. While laboratory testing is necessary for development work, field testing of actual installations in operation is considered the only acceptable method of determining performance, and, while still possessing many trouble spots, is gradualllly becoming standardized.

Electrostatic Air Cleaning as Applied to Air Conditioning

By J. R. McCONNELL†

SINCE the introduction of air filtration in 1921, progress in performance and design have kept pace with the requirements. But the possibilities of mechanical air cleaning as exemplified by present day viscous and dry type filters have been so thoroughly explored that further improvements will be in refinements rather than fundamentals. Efficiencies of modern air filters are entirely adequate for most air conditioning applications.

The real problem has been the tremendous increase in atmospheric pollution during the past decade. This increase has been so gradual that its extent has not been appreciated. As dust concentrations increased, air conditioning engineers insisted upon more efficient air cleaning. To complicate matters, outlet grilles are no longer just grilles, but are disguised as lighting fixtures, or become decorations, to be placed where they will be most inconspicuous. Air is permitted to blow where it listeth, which is usually across an acoustical ceiling or against a rough wall and filters are blamed for the resulting discoloration. No one stops to consider that thousands of cubic feet of air come in contact with that discolored area; that, while each cubic foot might contain relatively few dust particles, the total number impinged upon this spot would be a considerable amount. That the air moving rapidly over a roughened surface can create an electrostatic discharge which will actually precipitate the smoke and dust particles.

If electrical precipitation can produce this discoloration, then the proper application of this same principle should eliminate the smoke and fine dust particles that cause the trouble. The Cottrell method requires high potentials ranging from 30,000 to 100,000 volts and consumes appreciable current. Also it generates large volumes of ozone and nitrous oxide which makes its use impractical for space ventilation or air conditioning service. The high efficiency of electrical precipitation in removal of smoke and fine dust has long been recognized, but full credit must be given to G. W. Penney, research engineer of Westinghouse Electric & Manufacturing Company for adaptation of this principle to air cleaning service. It was he who separated ionization and collection into separate units which made possible the use of much lower voltages and reduced the ozone generation to a negligible quantity.

In this arrangement the ionizing unit consists of fine wires equally spaced between metal tubes. The wires carry a positive potential of approximately 13,000 volts and the tubes are grounded. This high voltage reducing the surrounding air molecules into negative electrons and positive ions, the wire, being positive, attracts the negative electrons and the positive ions travel at high velocity to the negatively charged or grounded tubes. Since this action takes place along the entire length of the wires, the constant flow of ions forms a continuous field or curtain of rapidly moving ions between the wires and electrodes.

Dust particles, passing through this field are bombarded by these positive ions. In normal atmosphere most dust particles have a balanced charge of negative electrons and positive ions. In passing through the ionizing unit they acquire a surplus of positive ions and become postively charged or ionized. As the ionized dust particles enter the electrostatic field between the parallel metal plates of the collector unit, closely spaced and alernately charged and grounded, they are repulsed by the positively charged plates and attracted to the negatively charged or grounded plates. This causes precipitation of the smoke and fine dust particles on the grounded plates.

With plates parallel to the air flow, their depth, in the direction of air flow is determined by the theoretical part of a dust particle in responding to the resultant forces of air velocity and the electrical attraction that tends to draw it to the negative plate. Thus the arrangement of the plates, their spacing and the air velocity between them have a direct bearing upon the percentage of dust removal.

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Once the dust particle strikes a grounded plate it gives up its positive charge and is no longer attracted to the plate except by its natural adhesiveness. Particles lacking this natural adhesiveness could be jarred loose from dry collector plates and carried along in the air stream. Dust may also be liberated by electric sparks caused by flash-over between the plates.

Both these conditions are eliminated or alleviated by a viscous coating on the plates. Another advantage of an oil film on the plates is that being a dialectric, it is possible to maintain a higher voltage on the plates, or to reduce the spacing between the plates, due to its insulating effect. This results in greater electrical attraction being exerted on the dust particle, which means that the depth of the plates can be reduced, or the velocity of air travel increased.

An interesting phenomenon in this connection is the leveling effect of high voltages on an oil film. It has been found that collector plates with the current on carry a more uniform oil film than when the current is off, and that the thickness of the film is definitely limited. The minute the current is applied to the freshly oiled plates all irregularities in the oil distribution disappear and the oil film becomes uniform over the entire plate area. Any surplus oil will immediately flow to the lowest spot on the plate and drain off. This action materially reduces the possibility of oil entrainment in a viscous coated electric precipitator.

The efficiency of the electrostatic method of cleaning air, particularly with the finer particles which cannot be impinged on microscopic slides or accurately weighed, has made it necessary to turn to another method of determination. Since its performance extends into the realm of smoke, the discoloration method has been found the only practical means of evaluating efficiency of electrostatic filters. In this method equal quantities of cleaned and dirty air are drawn through chemical filter paper. A comparison of the degree of discoloration after a given time, or the length of time required to produce the same degree of discoloration, gives an efficiency rating for the air unit.

It has been determined that an efficiency of 85% by the discoloration method is sufficient for most air conditioning service. Some applications may require 90%, which can be obtained by reducing the air velocity by approximately 20%. As a comparison, the efficiency of the best felt filters is only 25%.

Combining the Penney principle of standard electrostatic units with an automatic self-cleaning viscous filter, gives higher efficiencies over the entire range of particle sizes with the advantage of automatic operation. Another advantage of this combination is that in case of power failure, that might disrupt electrical precipitation, the viscous air filter would continue to function and provide a normal degree of air cleanliness.

The collector plates in this type are automatically cleaned at predetermined intervals by the rotation of the filter curtain through an oil bath. This action also renews the viscous film on the collector plates. Design of this filter curtain required considerable ingenuity, owing to the many limiting factors. It had to consist of parallel plates spaced approximately one-quarter inch apart. These had to be electrically insulated from each other and provided with means for applying electric potential of opposite polarity to adjacent plates. And the possibility of oil entrainment has been so completely minimized in the machine I am demonstrating that it is possible to reverse the direction of rotation of the filter curtain so that the front curtain with its heavier dust load is carried directly down into the oil reservoir, while the rear curtain is always clear, with a fresh viscous coating.

$100,000 Roadside Restaurant Conditioned

It is not often that $100,000 is spent on the construction and equipment of a roadside restaurant, but such is the case with the newly opened Smith House at 400 Memorial Drive, Cambridge, Massachusetts, past which thousands of automobiles drive daily. This is the last word in roadside eating places, with one of the most elaborate and complete kitchen layouts of any restaurant in the country, complete air conditioning for the main dining room, and air conditioning planned for smaller upstairs dining rooms.

The restaurant overlooks the Charles River and accommodates 350 guests in the main dining room which is enclosed on three sides by glass and looks out on a terrace dining space. There are three additional rooms on the second floor for special parties. Glass and silverware cases and a battery of storage refrigerators are equipped with sterilamps.

There are 20 tons of refrigeration for air conditioning the main dining room. Controls are of the compensating type, giving 72F to outside 75 or lower and progressive to 72 — 80F against 75 to 95F outside temperature.

Direct radiation is provided under the large windows and is controlled by a separate thermostat. The heating plant consists of a 95 hp. water tube boiler fired by a burner using heavy oil.

It is planned to install an additional air conditioning system for the second floor, using probably well water in place of compressors. A test well driven, however, brought in strongly salt water, although the Charles River, with only slightly brackish water, is only about 200 ft. away, so another well will have to be driven.

The restaurant was built by William E. Smith, a veteran in the restaurant business, as his conception of a model eating place, and combines outstanding features of food service and sanitation throughout.

EDITORIALS

Stability in Summer Equipment

Signs are discernible that we are entering a period of relative stability in the matter of summer air conditioning equipment. It is evident that now there is less certainty on the part of equipment makers as to their ability to crack the situation wide open with anything which they now have to offer or which they can produce in the immediate future. So also are dealers and installers of this equipment less certain than heretofore that there is a huge popular demand to be quickly turned into profits by the usual methods of approach. To many purchasers the available equipment has lost much of its former mystery and they are now in positions to appraise its cost, its performance, and its benefits to them and their businesses. Furthermore, it is being seen that market demand is by no means constant in all parts of the country or among all classes of prospective buyers. The market is selective as to both locality and application. These are all factors which tend toward stability, for with closer buyer scrutiny, with a questioning attitude on the part of the trade and with the manufacturers having already extended their designing and producing abilities the situation is not favorable to sudden fluctuation.

This need not be taken as indicating that the net prospects are any better or any worse than they have been heretofore. It merely means that some of the factors which have always been present in the situation are being better recognized and are being evaluated more realistically. It means that reason is replacing wishful thinking and that some of the romancing indulged in in the early days is being brought down to earth.

Individuals here and there appear to have lost faith in the future of summer cooling. This is inevitable. Also it is inevitable that the present stablization shall affect adversely some commercial operations. But so far as the general prospects are concerned there is much to be said in favor of a period of relative stability in regard to equipment for summer cooling since it has virtually run wild during the past few years. A reasonably stable period would permit moving of inventories, would let the distribution situation jell, and would let owners further decide what features they really want and what ones they can get along without. What is more, it would let the cooling phase of the air problem in buildings start to assume a justifiable position as related to the other phases of the same problem. This is a development greatly to be desired and one which is likely to interfere with the healthy commercial growth of the whole business unless and until it is straightened out. Let no one be misled, a period of stability is no indication that a future is lacking. Rather it is a good sign that the future of summer cooling is to be more definitely assured.

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A Test of Smoke Abatement

A revised and relatively severe anti-smoke ordinance has gone into effect in St. Louis carrying the proviso that fuel for hand-firing may not be used in the city if it contains more than 23% volatile material. If mechanical firing is used, however, this restriction does not apply. It is anticipated that one effect of the ordinance will be an increased use of mechanical stokers and of other means of mechanical firing such as automatic burning of gas and oil in building heating. Those directly interested in this phase of the situation have already moved aggressively to encourage this result. This may be regarded as a move on the part of those who are in favor of trying to make the new ordinances work and thus to rid the city of the worst of its smoke cloud.

This activity does not tell the whole story, however, for there are interests who would be well pleased if the operation of the ordinance were to show no improvement in the smoke condition throughout the city. These are generally those people who are adversely affected by the ordinance and who would be glad to see it repealed or not enforced. Just now they have no public program of opposition but can be expected to move quickly at a favorable opportunity. Their move would logically be an attempt to influence public opinion to regard the ordinance as a failure and as a nuisance.

Thus, as the situation now stands there is an advanced ordinance on the law books, there is a campaign under way looking toward helping it to work, and there is a latent opposition to go into action to neutralize the effects of the ordinance if the course of events should bring about a suitable opportunity.

To all engineers familiar with the fuel problem and with the many moves made at various times in a great many of our cities to abate the smoke nuisance, this situation in St. Louis holds much of interest. Probably this ordinance comes closer to putting teeth into a smoke law than has been true of any of the other ordinances. Engineers have generally agreed that the smoke problem is one which can be readily solved technically but whose actual solution depends almost entirely on securing a really effective degree of public support. A test of the soundness of this viewpoint under relatively favorable circumstances now looms up in St. Louis. It is worth watching carefully during the coming months.

ABSTRACTS of Current Papers, Books and Pamphlets

Air Conditioning Unit for Textile Tests

Equipment is described for the study of the moisture relations of materials in air of controlled relative humidities at elevated temperatures. The air is brought to the desired relative humidity and temperature by an arrangement including a saturator and automatically controlled heaters, and then passed through a working chamber in which the materials to be studied can be weighed rapidly and accurately. Provision is made for measuring the rate of flow of air and for accurately determining the relative humidity in the working chamber.

The equipment has been used at temperatures from 96F to 300F, at relative humidities for 10 to 90 per cent for temperatures up to 212F, and up to the maximum relative humidities obtainable at atmospheric pressure for temperatures between 212F and 300F. The temperature in the working chamber can be maintained constant within ± 0.4F and the relative humidity within ± 1 per cent relative humidity for experimental runs of several days' duration.

Data are given illustrating the performance of the equipment.

[*"Equipment for Conditioning Materials at Constant Humidities and at Elevated Temperatures," by James G. Wiegerink. Research paper RP1303, National Bureau of Standards. Six pages. Paper for sale by the Superintendent of Documents, Washington, D. C. Price, 5c.*]

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Air Conditioning

Since 1932 test work has been carried on at the University of Illinois Research Residence on various methods of summer cooling. This report describes the investigations made during the summers of 1937 and 1938 to determine the temperature and humidity conditions that could be maintained with a mechanical refrigerating unit operated at two different capacities, 18,000 and 30,000 B.t.u. per hour. Tests were made (1) with both floors of the residence cooled simultaneously, (2) with cooling on the first story during the day and on the second story at night, (3) with cooling on the second story only, and (4) on the heat flow through walls.

When both floors were cooled simultaneously with the 18,000 B.t.u. unit it was necessary to carry a higher relative humidity to maintain the required effective temperature than when the 30,000 B.t.u. unit was used. On days of excessive heat it was not possible to maintain the desired temperature with the smaller unit. However on days when the daily median temperature was lower than 83F (maximum outdoor temperature 95F and a minimum night temperature of 71F) the unit could be considered to give satisfactory performance. Studies of past weather records at Urbana. Illinois, indicated that since this condition is exceeded only during a relatively short period during the summer, the use of the smaller condensing unit is feasible for residences, especially in those cases in which the initial cost of installation is a more important factor than the maintenance of ideal conditions.

The tests run with the first story cooled during the day and the second story at night using the smaller unit, indicated that this method of cooling was only reasonably satisfactory. The comfort conditions maintained on the first floor were only slightly more satisfactory than with the same unit cooling the entire house. Comfort conditions on the second floor were on the borderline of comfort and due to the heat in the structure a considerable time was required to reduce the air temperatures to comfortable levels. The investigators pointed out that with this method of operation on mild days the on-off operation of the unit was not satisfactory due to a rapid rise in relative humidity and effective temperature during off periods. In this connection they mentioned that the use of a cooling unit or operating method which was just sufficient to maintain the desired indoor temperature at all times would eliminate these off periods and the accompanying periods of high humidity. Any method of control or operation which would tend to modulate the capacity of the cooling unit to more nearly conform with the cooling requirements, without increasing the relative humidity, should prove more satisfactory than the intermittent operation of a unit having a fixed capacity.

The results of tests with cooling on the second story only were quite satisfactory. With this method of cooling it was found that both stories were satisfactorily cooled inasmuch as the cold air from the upper story was drawn downstairs and returned to the air conditioning unit through return air grilles on the first floor. The investigators were careful to point out that the application of this principle may be limited by the arrangement of the first story, particularly with respect to the location of the return air grilles and the location of rooms remote from the stairway.

The studies of heat flow through the walls of the research residence indicated that the application of insulation to a frame wall materially reduced both the maximum hourly heat flow and the total heat flow through the wall per day. However, the reduction in both the maximum and total heat flow was less than that indicated by the ratio of the calculated coefficients of heat transmission of the uninsulated and insulated walls. It was also found that the application of insulation reduced the temperature of the inside wall surface thus resulting in conditions more conducive to comfort for a given air temperature than those produced by the uninsulated wall. The tests also showed that the insulated wall cooled to a temperature approximating that of the indoor air by 2 a.m., thus indicating that there was no material storage of heat in the insulation to be carried over to the following day and to become cumulative in the case of a succession of hot days.

[*"Summer Cooling in the Research Residence with*

a Condensing Unit Operated at Two Capacities," by A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick. Engineering Experiment Station Bulletin Series No. 321. Published by the University of Illinois, Urbana, Ill. Paper cover; 6 x 9 in.; 72 pages. Price, 70c. Copies available free until December 1, 1940 or until the supply available for free distribution is exhausted.]

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Fuels

Here is a book on fuels and their utilization which rigidly limits itself to its subject and does not attempt to cover any more of the science of thermodynamics than is indicated by its title. It takes up the various fuels which are available for production of heat, the means by which they are utilized for this purpose, and the modern investigations and tests to increase and maintain efficiency in the production of heat.

The book is divided into eleven chapters entitled as follows: Introduction; Combustion; Classification of Fuels, Gaseous Fuels — Liquid Fuels — Solid Fuels; Manufactured Fuels: Combustion of Coal and Other Solid Fuels; Waste Heat and Its Elimination—Storing of Coal; Calorimetry Determination of Heat Input; Heat Balances, Thermochemistry—Boiler Computations—Producer Calculations; Pyrometry or Temperature Measurements; Water for Industrial Purposes, and Laboratory Experiments.

At the end of each chapter there is a number of questions for use when the book is to be used as a textbook while in the back is a number of laboratory experiments which serve to make the book useful as a laboratory manual as well.

[*"Fuels and Their Utilization," by A. R. Carr and C. W. Selheimer. Published by the Pitman Publishing Corp., 2 West 45th St., New York. Cloth bound, 6 x 9½ in., 177 pages. Price $2.*]

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Air Infiltration

While this is the 45th report of the National Bureau of Standards on the results of their research program on building materials and structures, it is the first one of direct interest to the heating and ventilating engineer. The purpose of this particular research work was to develop equipment for conducting air infiltration tests at the National Bureau of Standards and to determine the relation between infiltration and sash clearances for commercially available windows of types suitable for use in low-cost housing.

The equipment developed consists essentially of a source for supplying air, a storage tank, an integrating meter for flow measurement, and a semi-automatic timing system. A special test-panel construction and sealing compound were developed and used to insure minimum spurious leakage during tests.

To simplify the presentation and use of the relation between infiltration and window pressure the investigators developed a method by which the results are expressed in terms of infiltration ratios. They define the infiltration ratio, at a given pressure for a window under specified clearance conditions, as the ratio of the infiltration of the window at that pressure to the infiltration at the reference pressure of 0.200 in. of water with the other conditions the same. It was found that for a large range of types and conditions of windows the curves of infiltration ratio plotted against the pressure were in approximate coincidence over a pressure range from zero to about 0.60 in. of water.

Values of the infiltration ratio obtained from this overall average curve are given in the table.

VALUES OF INFILTRATION RATIO

Pressure In. of Water	Wind Speed Miles per Hour	Infiltration Ratio
0.05	10.2	0.38
0.10	14.4	0.63
0.15	17.7	0.82
0.20	20.4	1.00
0.25	22.8	1.15
0.30	25.0	1.30
0.35	27.0	1.43
0.40	28.8	1.55
0.45	30.6	1.67
0.50	32.2	1.78
0.55	33.8	1.88
0.60	35.3	1.98
0.65	36.8	2.06
0.70	38.2	2.12

The data given in this table may be used within the limits of error usually acceptable in the design computation to estimate the infiltration for common windows at any pressures within the range of 0.60 in. of water, provided that infiltration at any other pressure within the range is known.

[*"Building Materials and Structures. Report BMS-45. Air Infiltration Through Windows" by E. F. Coleman and R. H. Heald. Published by the National Bureau of Standards, U. S. Department of Commerce. Paper cover, 8 x 10½ in. 22 pages. For sale by the Superintendent of Documents, Washington, D. C. Price 10 cents.*]

BRIEF REVIEWS

Coal. From 1923 to the present time, the Canadian Bureau of Mines has been conducting a series of burning tests in order to determine the comparative value of central Canadian fuels with a standard American anthracite fuel when burned in a domestic hot water boiler. In this report a comprehensive summary of all the burning tests made in these laboratories on domestic coals and domestic coal substitutes is presented. [*"Comparative Tests of Various Fuels when Burned in a Domestic Hot Water Boiler," by C. E. Baltzer and E. S. Malloch. Published by the Bureau of Mines, Mines and Geology Branch of the Department of Mines and Resources, Ottawa, Canada. Paper cover; 6½ x 10 in.; 23 pages, together with a number of large tables. Price, 25 cents.*]

Stokers. A standard of the Stoker Manufacturers Association recommending minimum setting heights for stokers. Includes a large size diagram giving the minimum setting height when the burning rate is known. [*"S.M.A. Minimum Setting Heights." Available from Marc G. Bluth, Secretary, Stoker Manufacturers Association, 307 North Michigan Avenue, Chicago, Ill. Price, 10 cents, postage prepaid.*]

NEWS OF THE MONTH

MIT Studies Service Water Use

BOSTON—How much oil is needed in the summer months to run an indirect water heater for domestic hot water supply in the average home is a question that has puzzled many oil burner men and burner owners. According to Professor Gordon B. Wilkes, Massachusetts Institute of Technology, it requires 85 gallons of oil per month when producing the typical 50 gallons needed daily for the average family.

Applying this to current prices for oil at 6½ cents a gallon, this amounts to $22.50 for the four summer months. According to Prof. Wilkes, it shows that there is little economy in operating an indirect water heater in the summer time. This oil use is seldom realized, as it generally represents left-over quantities in the storage tank. But the cost remains, nevertheless.

Winter operation of an indirect water heater is much more economical, requiring perhaps only one-quarter of the oil that is used in summer. This is because the boiler operates anyway, and the heat for the water heater is simply a by-product. Under typical conditions the summer time efficiency will be between 10 and 15%, whereas the same boiler heating the house and water will have an efficiency of 55%.

Burner Men Discuss Trade Problems

NEW YORK — Fifty-one individuals representing 30 oil burner manufacturers and eight accessory manufacturers met at Eastern Point July 8 at the invitation of Oil Burner Institute for a conference on mutual industry problems and objectives. Also in attendance were representatives of Federal Housing Administration, Underwriters' Laboratories and The Institute of Boiler and Radiator Manufacturers. The Commercial Standard CS75-39 was widely discussed and several requests were made from the floor for its wider adoption by FHA. It is understood that this will be forthcoming at an early date.

R. E. Ferry, general manager of The Institute of Boiler and Radiator Manufacturers, explained to the meeting the development of the I-B-R ratings recently established and expressed belief that these standards and the oil burner standards in conjunction, would be a strong factor in promoting better installations and increased consumer satisfaction.

Trade practices for the oil burner industry were discussed broadly by C. E. Lewis, general sales manager of Delco Appliance Division, General Motors Sales Corporation, and the legislative situation was canvassed by W. A. Matheson of the same company.

Fluorescent Lighting Institute Touches on Air Conditioning

CAMBRIDGE, MASS. — New England's first Fluorescent Lighting Institute was held at Massachusetts Institute of Technology July 9-10, with a large attendance of illuminating and air conditioning engineers, salesmen, contractors, architects, wiring inspectors and others with illuminating problems. Every attempt was made to have the lectures and papers as practical as possible, with theory minimized and delivery in as non-technical language as possible. In consideration of the many present who were comparatively unfamiliar with the technicalities of electrical illumination, the talks, in some cases, were more or less elemental. The address by P. N. Clerke, illuminating engineer of the Westinghouse Electric Company, Boston, on "Fluorescent Lighting and Air Conditioning" was of interest to air conditioning engineers.

Mr. Clerke said in part: "The lighting engineer has given very little, if any, consideration to the subject outside of the correct foot candle on the working plane; the air conditioning engineer, on the other hand, has been prone to look upon lighting as just so many watts per square foot and another bothersome heat gain to overcome.

"Of the total electrical energy supplied by any of the various types of incandescent or mercury vapor lamps, only approximately 11% is turned into light and the remaining 89% is heat, of which 78% is invisible, in the infra-red end of the spectrum. The fluorescent lamp seldom has a heat rise of higher than the ambient room temperature. Of the total electrical energy supplied to this type of lamp, 22.5% is turned into useful light and the remaining 77.5% is heat, of which 43.5% is in the infra-red region.

"It can readily be seen that the fluorescent lamp is more desirable to work with from the air conditioning engineer's viewpoint. It does have, however, its limitations, both from the illumination and air conditioning standpoints. Being a gaseous discharge type of lamp, the light emitted in a draft of air will fall very rapidly. In the case of a base or exposed lamp this may vary in relative light output from 10 to 90%, with temperatures ranging from zero to 75F. If the lamp is enclosed, in a fixture, with glass around it, this relationship changes in the order of 60 to 90% relative light output with temperatures varying from zero to 60F. These figures are only relative and give only the approximate limitations."

Hatch Joins Univ. of Penn. Staff

PHILADELPHIA—Theodore Hatch, nationally known for his work in connection with industrial dust and dust control, has been made Associate Professor of Industrial Hygiene in the Department of Public Health and Preventive Medicine, University of Pennsylvania Medical School.

Prof. Hatch is a graduate of the University of Maine in Civil Engineering and also of the Harvard Engineering School in Sanitary Engineering, and was formerly instructor in industrial sanitation at Harvard. Until his acceptance of the position at the University of Pennsylvania he was Associate Dust Control Engineer with the Division of Industrial Hygiene, New York Department of Labor.

Prof. Hatch is the co-author with Prof. Philip Drinker of Harvard, of the book "Industrial Dust" and is the author of a number of technical papers among which have been various articles in HEATING & VENTILATING.

Resigns from Warm Air Association

COLUMBUS — Twenty-seven years of service as the managing director and treasurer of the National Warm Air Heating and Air Conditioning Association was completed July 1 by Allen W. Williams, when he voluntarily retired. Mr. Williams, however, is not severing his connection with the organization completely. He will continue with the Association in an advisory capacity, giving what advice and suggestion he can if and whenever needed.

He also continues as secretary and treasurer of the National Warm Air Register Manufacturers Institute, a position he has held for many years.

Mr. Williams is succeeded as secretary and treasurer of the National Warm Air Heating and Air Conditioning Association by George Boeddener who has been serving the same organization for some time as assistant to the president.

Senior Kahn Men Now Stockholders

DETROIT — In recognition of the efforts of members of the firm who have been associated with the organization for more than 25 years a new corporation has been formed by Albert Kahn, Inc., making those men with long service records stockholders in the company.

The new firm will be known as Albert Kahn Associated Architects and Engineers, Inc., with Albert Kahn and Louis Kahn heading the organization. Albert Kahn, Inc. will complete the work it now has on the boards but all future commissions will be undertaken by the new corporation.

Corrosion Group Elects Speller

ATLANTIC CITY — The second annual meeting of the American Coordinating Committee on Corrosion was held here June 27. Dr. F. N. Speller, Pittsburgh, consultant on corrosion, was formally named chairman for the coming year. Dr. R. M. Burns, Bell Telephone Laboratories, was named vice-chairman and Dr. G. H. Young, Mellon Institute of Industrial Research, secretary-treasurer. Committee headquarters are 4400 Fifth Avenue, Pittsburgh.

The Committee was organized two years ago to coordinate research activities in this extremely important field, and is patterned after similar organizations abroad. Cooperation has been promised by major companies and laboratories actively engaged in corrosion investigations.

The Committee is at present composed of official delegates from American Chemical Society, American Electroplaters Society, American Foundryman's Association, American Gas Association, American Institute of Chemical Engineers, American Institute of Electrical Engineers, American Institute of Mining and Metallurgical Engineers, American Society of Heating and Ventilating Engineers, American Society of Mechanical Engineers, American Society of Refrigerating Engineers, American Society for Metals, American Society for Testing Materials, American Water Works Association, Battelle Memorial Institute, Electrochemical Society, Mellon Institute of Industrial Research, National Bureau of Standards, National District Heating Association, Society of Automotive Engineers, and Technical Association of Pulp and Paper Industry.

ASHVE Schedules Fall Meeting

NEW YORK—Dr. F. E. Giesecke, College Station, Texas, president of the American Society of Heating and Ventilating Engineers, announces that the first Society Meeting ever held in Texas will be the 1940 Fall Meeting, October 14-15 at Houston, Tex., with the South Texas chapter as host. The meeting will be a two-day session at which six technical papers will be presented. Headquarters for the meeting will be at the Rice Hotel.

Arrangements for the meeting and the entertainment of visiting members will be directed by C. A. McKinney, general chairman, and he will be assisted by the following committee chairmen: J. V. O. Weaver, vice-chairman; J. A. Walsh, publicity; W. R. Etie, transportation; I. A. Naman, tours; A. M. Chase, Jr., banquet; D. S. Cooper, registration; A. B. Banowsky, entertainment; R. F. Taylor, finance; A. F. Barnes, ladies; G. D. Maves, sports.

Shows Flowers Stored for Months in Low Temperature Rooms

MOBILE, ALA. — In addressing the Mobile Rotary Club July 19, A. A. Richards, manager of the cold storage unit of the Alabama State Docks here, reported he is conducting an experiment whereby flowers may be stored for several months, remaining in a dormant state until they are exposed to normal temperatures.

On the speaker's rostrum was a bunch of flowering gladioli, which Mr. Richards said had been taken from the storage plant the day before. He also produced a wrapped bunch of the flowers, which he said had been taken from storage only a few hours before, and pointed out that these flowers were exactly as they had come from the field, with buds just beginning to burst. Richards said the same experiment was being carried on with peonies, and as far as he could see was an apparent success.

The speaker explained how flowers, particularly gladioli, which grow abundantly near Mobile, could be stored for several months and placed on the market when they would bring the best price.

Exhibiting containers of frozen shrimp and fish, Richards said the plant has a quick freeze system which can freeze seafoods to 0F in 15 minutes. He declared this is the only method by which foodstuffs may be stored for lengths of time without losing some part of the food value of the product. Strawberries, butterbeans, green peas and asparagus were also exhibited and passed around for Rotary members to view first hand, each article being frozen hard as ice.

Mr. Richards noted the rising importance of cold storage plants, saying they have made their most progress in the past 10 years. The process of cold storage is about 35 years old, he said, and plays a most important part in the event of a war or national emergency. Citing a tremendous figure representing the amount of meat now in storage in plants all over the nation, Richards said this meat would be sufficient to feed the armies of Europe for only one week. In the event of an emergency the nation's plants could be used to store reserve foodstuffs.

The Mobile cold storage plant houses about 20 major products, Richards said, while hundreds of other articles are also in its lockers.

New U.S.Liner America Conditioned

NEW YORK—The new S.S. America, the largest passenger liner ever to be constructed in this country and which arrived in New York Monday, July 29, is extensively air conditioned.

OBI to Hold Burner Show in Phila.

NEW YORK—Oil Burner Institute announces that an oil burner show will be held at Philadelphia in the spring of 1941. Any exhibit designed to illustrate the development of the oil burner industry and of oil burning equipment may be shown including displays of everything entering into the manufacture, installation and use of oil burners. This will include the adaptation of equipment for heating, cooking, and power, as well as air conditioning. Final plans for the show which will be open to the public, will be approved by the Executive Committee in a few weeks, according to C. F. Curtin, Institute Secretary. A convention will be held coincident with the show, as well as the annual meetings.

The Institute is offering a cash prize of $50 to members of the industry for a name for the show. Suggested names, to be considered, must be received at Institute headquarters, 30 Rockefeller Plaza, New York City, not later than August 20, and the winning name will be selected by the executive committee whose decision will be final. Names will be recorded in the order of receipt and should more than one contestant submit the winning name, the prize will be awarded to the first submission received. Members of the industry are invited to participate in the contest which is not open to any officer or director of the Institute.

Joseph L. Baker

OMAHA—Joseph L. Baker, a founder of the Baker Ice Machine Company and the United States Gypsum Company, died here June 27. He was 85. He had been inactive the last two years and had been in the hospital since June 22.

Mr. Baker discovered that gypsum rock could be used in plaster to make a hard and waterproof finish, so that in 1888 with two associates he bought a gypsum mountain in Blue Rapids, Kansas, and later other gypsum properties. He was on the original executive committee of the United States Gypsum Company, which was later sold to the National Gypsum Company.

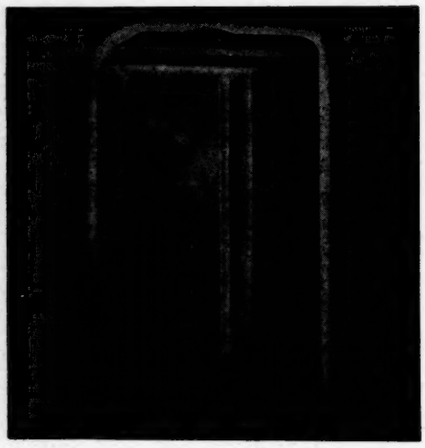
The Baker Ice Machine Company, which he started with a partner in 1905, and acquired as sole owner in 1907, was started in a small shop in a one-story building at Thirteenth and Howard Streets in Omaha, Nebraska. Mechanical refrigeration became Mr. Baker's major interest and most of his time and effort were devoted to working out new ideas and developing foreign markets for the equipment.

Surviving are two sons, Richard L., Omaha, and Chester A., New York; a daughter and seven grandchildren.

NEW EQUIPMENT

Penn Thermostat

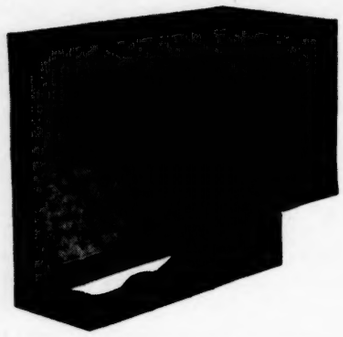
NAME—Penn Thermostat 870 Series.
PURPOSE—For controlling room temperature.
FEATURES—A two-wire, low voltage, heat anticipating or auxiliary heat actuated thermostat which replaces the company's former type 836 thermostat. The following advantages are claimed: a one-piece moulded bakelite base designed for three-point mounting; terminal set in recesses moulded into the base to eliminate possibility of shorting across the terminals; the use of a powerful, though small, Alnico magnet in the permanent magnet snap-acting contact; and a convenient sealed calibration adjustment. The thermostat is designed for use with all of Penn's small motor relays and primary controls for oil, gas, and stoker heating service. 870 Series also available for automatic day-nite control.
MADE BY—*Penn Electric Switch Co., Goshen, Ind.*



For comfort heating applications.

McDonnell Humidifying Valve

NAME — McDonnell humidifier water control.
PURPOSE—For controlling the flow of water to humidifier pans.
FEATURES—Control features a snap-action valve which is designed to eliminate valve sticking or clogging. A cam and roller action causes the valve to snap from tight closure to wide open whenever the water level drops one quarter of an inch. Designed for use against water pressure up to 150 lbs. Made in three styles: No. 17, valve and float only; No. 117, valve and float in covered float chamber; No. 217 includes copper tubing and a means of tapping into the water main.
MADE BY—*McDonnell & Miller, Wrigley Bldg., Chicago, Ill.*



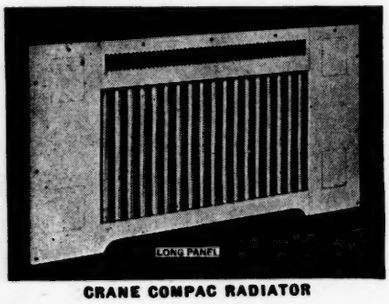
For balancing forced air systems.

Controlair Balancing Damper

NAME—Mayn Air Damper.
PURPOSE—For balancing forced air and air conditioning systems.
FEATURES—The damper is said to provide for accessible and rapid one-man operation of balancing a system which has registers and grilles already installed. It is for use in the stack-head and replaces the basement pipe dampers. Manufacturer states that an outstanding feature of this damper is that while it is readily accessible it is tamper-proof due to a special patented type of screw head which can be operated only with the use of a special tool supplied with it. Other features are said to be a permanent automatic locking device, positive control, and a rattleproof and vibrationless operation.
MADE BY—*Controlair, Incorporated, Elyria, Ohio.*

Crane Compac Radiator

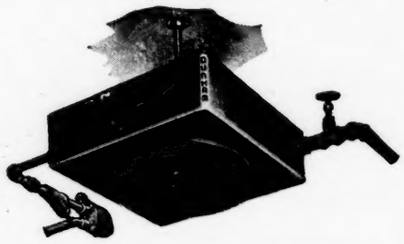
NAME—Crane Compac radiator.
PURPOSE—For room heating using steam or hot water.
FEATURES—Radiator is of slender tube type and is made in many sizes. It is available with either 3, 4, 5, and 6 tubes with bottom or end connections and may be either free standing or wall hung. Available with or without panels where radiators are recessed.
MADE BY—*Crane Co., 836 S. Michigan Ave., Chicago, Ill.*



CRANE COMPAC RADIATOR

Dunham Downflo Type Unit Heater

NAME AND MODEL NUMBER — Downflo unit heater, type C.
PURPOSE—For supplying a large volume of heated air at a relatively high velocity and low temperature vertically downwards to working levels.
FEATURES—This unit heater is designed for use in applications such as factories with high ceilings, factories where high clearances are necessary for clearing industrial equipment such as cranes and for over-doorway heat requirements of stores and office buildings. Heater is designed so that the motor and fan assembly may be removed for inspection or service without the necessity of removing the heater from the hanger or the disconnecting of the pipes. Casing is of heavy furniture metal. Motor, with the fan mounted directly on its shaft, is attached to spider members which grip a special resilient mounting which, in turn, is mounted on a conical air guide. Fan blades are said to



Dunham type C Downflo unit heater.

be of improved design set and shaped to the proper pitch and are designed for subdued air noise. Heaters are arranged for central support from a single hanger by a formed spider. A hole in the spider accommodates the pipe which serves as a hanger. Heating element is designed for operating at steam pressures up to 125 lb. and is made entirely of non-corrosive material. The air volume and temperature adjustment are provided by means of adjustable dampers. It is said that adjustment is readily made and retained in place. A choice of either top or side inlet is available for ease in connecting the steam piping. When properly installed, these heaters may be used on forced hot water systems.
SIZES AND CAPACITIES—Ten sizes with capacities ranging from 25,700 to 468,000 B.t.u. per hour based on steam at 2 lb. gage and 60F entering air. Air velocities range from 440 c.f.m. to 7000 c.f.m. Hot water capacities from 23,100 to 421,000 B.t.u. per hr. with 200F average water and 60F entering air.
MADE BY—*C. A. Dunham Company, 450 East Ohio St., Chicago, Ill.*

COMING EVENTS

AUGUST 15-17. The Second Annual Air Conditioning Short Course, at the Agricultural and Mechanical College of Texas, College Station, Texas. Sponsored by the Department of Mechanical Engineering.

SEPTEMBER 18-20. 18th National Industrial Advertisers Association Exposition to be held at Hotel Statler, Detroit, Mich.

OCTOBER 7-10. 22nd Annual Convention of the American Gas Association, to be held at The Hotel Traymore, Ritz-Carlton Hotel, Hotel Ambassador and the Atlantic City Auditorium at Atlantic City, N. J. Managing Director, Alexander Forward.

OCTOBER 8-11. The 69th annual meeting of the American Public Health Association, Detroit. Book-Cadillac Hotel, Headquarters.

OCTOBER 14-15. Fall Meeting of the Houston ASHVE at Houston, Texas, to be held under the auspices of the South Texas Chapter and with the North Texas Chapter Members cooperating.

DECEMBER 2-7. 14th National Exposition of Power and Mechanical Engineering at the Grand Central Palace, New York, N. Y. For information write National Exposition Co., Grand Central Palace, New York, N. Y.

JANUARY 13-16, 1941. Third Refrigeration and Air Conditioning Exposition to be held at Stevens Hotel, Chicago.

JANUARY 27-29, 1941. 47th Annual Meeting of the American Society of Heating and Ventilating Engineers to be held at Hotel Muehlbach, Kansas City, Missouri, with Kansas City as Host Chapter.

H&V'S PHOTOS

Pages 31-32—Hedrich-Blessing Studio for the Bankers Life Co., Des Moines, Iowa.

Page 36—American Society of Heating & Ventilating Engineers, New York.

WITH THE MANUFACTURERS

The Bryant Heater Co., Cleveland, and *Koppers Company*, Pittsburgh, have completed an agreement which links these two companies in an endeavor to establish a coke stoker as a major piece of equipment in automatic heating. For many years, Koppers has been working on a coke stoker. More recently it has given consideration to selecting an organization capable of manufacturing and selling the coke stoker and continuing further research and development work on future models. The Bryant Heater Company was selected because of the capabilities of its factory and also because of the past experience of the Bryant sales organization in running large scale merchandising campaigns for the successful selling of automatic heating equipment.

Manning, Maxwell & Moore, Inc., Bridgeport, Conn., has changed the name of one of its divisions from the Ashcroft American Gauge Division to Ashcroft Gauge Division.

Research Products Corp., Madison, Wis., has appointed *Allied Heating Products Company*, 225 Brokers Exchange Building, Norfolk, Va., as its agent for the State of Virginia to handle the distribution of the Research (Walton) air filters. *J. M. Stokley* is the President of Allied Heating Products Company.

Young Radiator Company, Racine, Wis., has appointed two new representatives as follows: *Roy M. Scott*, 417 Market St., San Francisco, to cover northern California; and *I. E. Danford*, 827 West St., Caldwell, Ohio, to cover southern Ohio.

RIC-WIL Insulated PIPE UNITS

—a speedy installation for Rochester Gas & Electric Corp.

THIS underground steam line, serving the Seneca Hotel, Rochester, N. Y., had to be installed completely between Saturday night and Monday morning, under a heavy traffic lane. This was accomplished, including back-filling and repaving, in 23 hours total elapsed time!

If, in addition to engineering dependability and long life, you want SPEED and ECONOMY—Ric-wiL Insulated Pipe Units can serve you too. Complete factory assembled units, in any lengths, with pipe covering as specified, and all necessary fittings, steam pipes furnished by customer or by us.

Send for Bulletin No. 4003.

The Ric-wiL Co., Union Commerce Bldg., Cleveland, O.

New York San Francisco Chicago

Agents in principal cities

REGISTERED IN U. S. PATENT OFFICE

RIC-WIL

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

NEW TRADE LITERATURE

Air Conditioners. A small pocket sized bulletin No. 521-A, entitled Frick Unit Air Conditioners, which shows typical installations, gives the features of construction, and information on the method of operation. The bulletin also contains some information on Frick refrigerating machines used in air conditioning work. FRICK CO., Waynesboro, Pa.

Controls. A standard size, 26-page catalog K, illustrating and describing the complete Julien P. Friez line of controls and instruments for automatic heating, refrigerating and air conditioning applications. Separate sections cover humidity and temperature controls for heating; temperature and pressure controls for refrigeration; and temperature, humidity, pressure, velocity and other measuring and recording instruments. JULIEN P. FRIEZ & SONS, Division of Bendix Aviation Corp., Baltimore, Md.

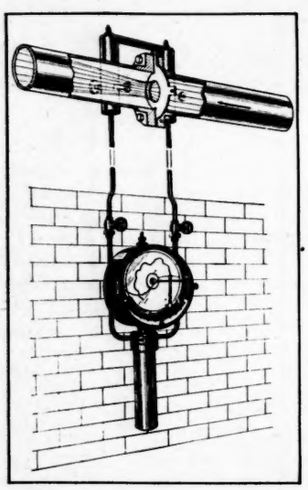
Convectors. A standard size, 24-page, four-color convector bulletin, S-380, illustrating and describing the Trane convector line for 1940. Bulletin is available in two editions—one with a French fold cover for presentation by architects, engineers, and contractors to their preferred clients, and the other with a regular trimmed flush cover punched for the convenience of Trane binder holders. Four-color views illustrate types of convector applications in residences, theaters, shops, offices and schools. Numerous other illustrations of applications in a wide variety of buildings are shown. Includes information on the advantages of convector heating, method of installation and styles available. THE TRANE COMPANY, La Crosse, Wis.

Fan Blades. An attractive, two-color 60-page catalog and instruction manual designed to simplify the selection of propeller fan blades. The first four pages are devoted to a description of the method of selecting fan blades, explanation of fan test codes, and a discussion of simple fan laws. The text of the catalog is made up of a series of bulletins on the various fan blades in the Torrington line. Each bulletin describes a particular type of fan blade and gives detailed specifications, performance characteristics, features of construction, and fan dimensions. Performance characteristics are presented in a graph and tabular form and show air deliveries and power consumptions as recorded under NAFM and NEMA code tests. The catalog is arranged for rapid reference and contains answers to all ordinary questions regarding propeller fan specifications and performance. THE TORRINGTON MANUFACTURING COMPANY, Torrington, Conn.

Water Heaters. A 12-page, three-color, standard size catalog No. 94, illustrating and describing the Kewanee storage water heaters. Includes full specifications and dimensions, conversion tables for the calculation of coil capacities per hour for various steam pressures with variable inlet and outlet water temperatures, the determinations of hot water loads and storage requirements, and the sizing of steam lines. KEWANEE BOILER CORPORATION, Kewanee, Ill.

The ADSCO FLOW METER, Orifice Type

ACCURATELY METERS STEAM, WATER, COMPRESSED AIR OR GAS.
SIMPLE IN CONSTRUCTION - - EASILY INSTALLED



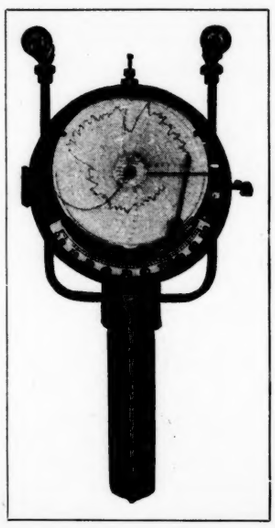
Free-Floating, frictionless Meter Mechanism transmits flow to direct-reading, evenly-divided chart.

The ADSCO Flow Meter has a record of *exceptional accuracy* at all rates of flow. It is highly sensitive to fluctuations in flow but cannot be damaged by sudden overloads or reverse flows. Simple and rugged in construction—easily installed and maintained—requires no frequent inspection or adjustments.

Available with recording chart, indicating scale and integrator counter to totalize the flow or in other combinations of these three devices.

Correspondence is invited regarding your metering problems so that ADSCO may make recommendations based on over 25 years' experience in the manufacture of meters.

Write for Bulletin No. 35-83V



AMERICAN DISTRICT STEAM COMPANY
NORTH TONAWANDA, N.Y.
IN BUSINESS OVER SIXTY YEARS

Degree-Day Figures for June, 1940

HEATING & VENTILATING continues its twelfth year of publishing degree-day data for various large cities

	Albany, N. Y.	Atlanta, Ga.	Baltimore, Md.	Birmingham, Ala.	Boston, Mass.	Buffalo, N. Y.	Burlington, Vt.
Degree-days for June, 1940	87	0	2	0	75	106	115
Degree-days, Sept. 1, '39 to June 30, '40	7719	3614	4838	3219	6457	7382	8515
Degree-days, Sept. 1, '38 to June 30, 39	7000	2594	4044	2249	5984	6709	8048
Degree-days, Sept. 1, to June 30, Normal	6580	2890	4533	2352	6045	6822	7514

	Cheyenne, Wyo.	Chicago, Ill.	Cincinnati, Ohio	Cleveland, Ohio	Columbus, Ohio	Denver, Colo.	Des Moines, Iowa
Degree-days for June, 1940	102	52	9	42	15	38	0
Degree-days, Sept. 1, '39 to June 30, '40	6883	6600	5577	6503	5934	5313	6604
Degree-days, Sept. 1, '38 to June 30, 39	7314	5840	4543	5491	4961	5637	5856
Degree-days, Sept. 1, to June 30, Normal	7466	6290	4703	6155	5398	5874	6384

	Detroit, Mich.	Dodge City, Kan.	Duluth, Minn.	El Paso, Tex.	Erie, Pa.	Evansville, Ind.	Fort Wayne, Ind.
Degree-days for June, 1940	40	9	230	0	58	0	16
Degree-days, Sept. 1, '39 to June 30, '40	6947	5062	9105	2382	6784	4980	6692
Degree-days, Sept. 1, '38 to June 30, 39	6246	4583	9487	2504	5884	3918	5736
Degree-days, Sept. 1, to June 30, Normal	6490	5035	9443	2428	6273	4244	5925

	Fort Worth, Tex.	Grand Rapids, Mich.	Green Bay, Wis.	Harrisburg, Pa.	Hartford, Conn.	Indianapolis, Ind.	Ithaca, N. Y.
Degree-days for June, 1940	0	46	85	18	77	7	75
Degree-days, Sept. 1, '39 to June 30, '40	2527	6796	7789	5813	6731	5849	7221
Degree-days, Sept. 1, '38 to June 30, 39	2168	6136	7406	5130	5882	4862	6491
Degree-days, Sept. 1, to June 30, Normal	2148	6535	7825	5375	6036	5298	6719

	Kansas City, Mo.	Knoxville, Tenn.	La Crosse, Wis.	Lansing, Mich.	Lincoln, Neb.	Little Rock, Ark.	Los Angeles, Calif.
Degree-days for June, 1940	5	1	19	58	8	0	11
Degree-days, Sept. 1, '39 to June 30, '40	5374	4168	7316	7325	6240	3517	739
Degree-days, Sept. 1, '38 to June 30, 39	4475	3286	6909	6651	5303	2789	1270
Degree-days, Sept. 1, to June 30, Normal	4852	3670	7322	7048	5999	2811	1504

	Louisville, Ky.	Madison, Wis.	Memphis, Tenn.	Milwaukee, Wis.	Minneapolis, Minn.	Nashville, Tenn.	New Haven, Conn.
Degree-days for June, 1940	2	24	0	78	29	1	57
Degree-days, Sept. 1, '39 to June 30, '40	5076	7428	3660	7087	8037	4222	6262
Degree-days, Sept. 1, '38 to June 30, 39	4010	6844	2761	6495	7753	3225	5586
Degree-days, Sept. 1, to June 30, Normal	4180	7429	2950	7245	7850	3507	5895

	New Orleans, La.	New York, N. Y.	Norfolk, Va.	Oklahoma, City, Okla.	Omaha, Neb.	Peoria, Ill.	Philadelphia, Pa.
Degree-days for June, 1940	0	31	0	1	6	2	6
Degree-days, Sept. 1, '39 to June 30, '40	1594	5741	3852	3750	6507	6244	5303
Degree-days, Sept. 1, '38 to June 30, 39	1008	4911	2878	3271	5605	5494	4531
Degree-days, Sept. 1, to June 30, Normal	1024	5347	3350	3613	6131	6109	4855

	Pittsburgh, Pa.	Portland, Me.	Portland, Oreg.	Providence, R. I.	Reading, Pa.	Reno, Nev.	Richmond, Va.
Degree-days for June, 1940	21	150	40	58	16	30	1
Degree-days, Sept. 1, '39 to June 30, '40	5809	7465	3435	6400	5687	4611	4458
Degree-days, Sept. 1, '38 to June 30, 39	4866	7070	3921	5738	4966	5216	3516
Degree-days, Sept. 1, to June 30, Normal	5235	7012	4469	6015	5389	5892	3727

	Rochester, N. Y.	St. Louis, Mo.	Salt Lake City, Utah	San Francisco, Calif.	Scranton, Pa.	Seattle, Wash.	Spokane, Wash.
Degree-days for June, 1940	66	0	31	178	57	57	41
Degree-days, Sept. 1, '39 to June 30, '40	7141	5091	4574	2184	6766	3536	5418
Degree-days, Sept. 1, '38 to June 30, 39	6466	4135	5421	2762	5913	4247	5874
Degree-days, Sept. 1, to June 30, Normal	6732	4585	5555	2876	6129	4883	6355

	Springfield, Ill.	Syracuse, N. Y.	Toledo, Ohio	Trenton, N. J.	Utica, N. Y.	Washington, D. C.	Wichita, Kan.
Degree-days for June, 1940	1	64	28	20	77	6	6
Degree-days, Sept. 1, '39 to June 30, '40	5725	7279	5788	5787	7855	4947	4892
Degree-days, Sept. 1, '38 to June 30, 39	4882	6434	5878	4986	7064	4099	4217
Degree-days, Sept. 1, to June 30, Normal	5373	6893	6077	4933	6796	4626	4673

Methods of applying the degree-day in connection with the operation of heating plants or for the prediction of fuel consumption of heating plants are described fully in the new enlarged edition of the DEGREE-DAY HANDBOOK, published by HEATING & VENTILATING, 148 Lafayette Street, New York, price $3. Degree-days as given above for a "normal" month or season are based on averages for a long period of years, ending about 1922. Averages covering different periods will disagree with the above figures slightly.

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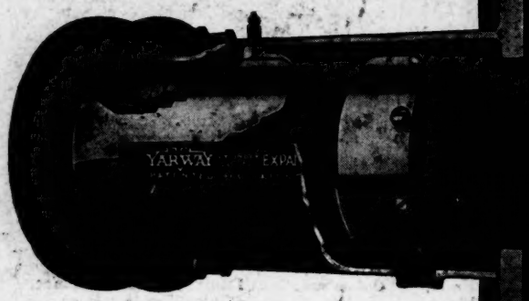

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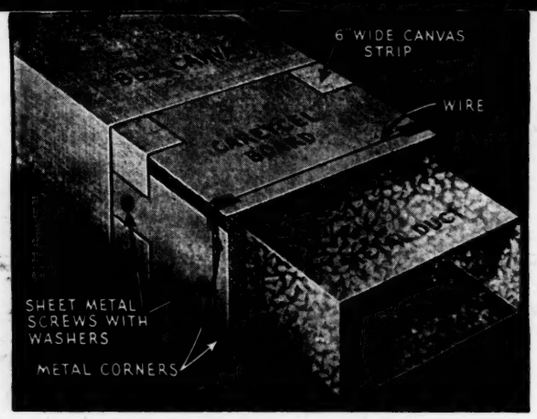

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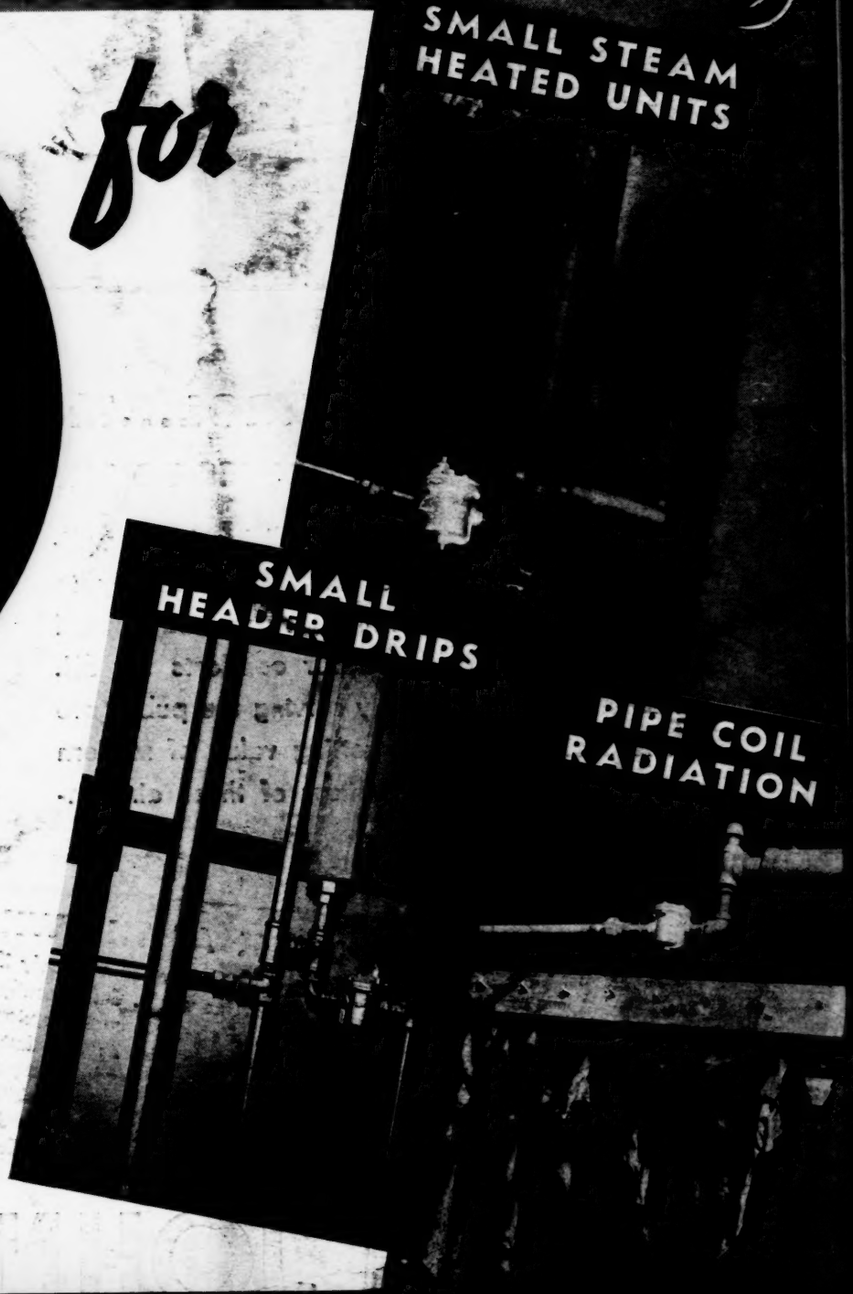

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